Cyclopedia
of
Civil Engineering

A General Reference Work on

SURVEYING, HIGHWAY CONSTRUCTION, RAILROAD ENGINEERING, EARTHWORK, STEEL CONSTRUCTION, SPECIFICATIONS, CONTRACTS, BRIDGE ENGINEERING, MASONRY AND REINFORCED CONCRETE, MUNICIPAL ENGINEERING, HYDRAULIC ENGINEERING, RIVER AND HARBOR IMPROVEMENT, IRRIGATION ENGINEERING, COST ANALYSIS, ETC.

Prepared by a Corps of

CIVIL AND CONSULTING ENGINEERS AND TECHNICAL EXPERTS OF THE HIGHEST PROFESSIONAL STANDING

NINE VOLUMES

AMERICAN TECHNICAL SOCIETY
CHICAGO
1921

Authors and Collaborators

FREDERICK E. TURNEAURE, C. E., Dr. Eng.

Dean of the College of Engineering, and Professor of Engineering, University of Wisconsin
Member, American Society of Civil Engineers
Joint Author of "Principles of Reinforced Concrete Construction," "Public Water Supplies," etc.

FRANK O. DUFOUR, C. E.

With Stone and Webster, Boston, Massachusetts
Formerly Structural Engineer with Interstate Commerce Commission
Formerly Assistant Professor of Structural Engineering, University of Illinois
Member, American Society of Civil Engineers
Member, American Society for Testing Materials

WALTER LORING WEBB, C. E.

Consulting Civil Engineer
Member, American Society of Civil Engineers
Author of "Railroad Construction," "Economics of Railroad Construction," etc.

W. G. BLIGH

Inspecting Engineer of Irrigation Works, Department of Interior, Canada
Formerly in Engineering Service of His Majesty in India
Member, Institute Civil Engineers (London)
Member, American Society of Civil Engineers
Member, Canadian Society of Civil Engineers

ADOLPH BLACK, C. E.

Civil and Sanitary Engineer, General Chemical Company, New York City
Formerly Adjunct Professor of Civil Engineering, Columbia University

EDWARD R. MAURER, B. C. E.

Professor of Mechanics, University of Wisconsin
Joint Author of "Principles of Reinforced Concrete Construction"

AUSTIN T. BYRNE

Civil Engineer
Author of "Highway Construction," "Materials and Workmanship"

A. MARSTON, C. E.

Dean of Division of Engineering and Professor of Civil Engineering, Iowa State College
Member, American Society of Civil Engineers
Member, Western Society of Civil Engineers

De WITT V. MOORE

Consulting Engineer and Architect
Formerly District Engineer—Central District Division of Valuation
Interstate Commerce Commission, Chicago
Member, American Society of Engineering Contractors
Member, Indiana Engineering Society

W. HERBERT GIBSON, B. S., C. E.

Civil Engineer
Designer of Reinforced Concrete

JAMES K. FINCH, C. E.

Associate Professor of Civil Engineering, and Director of Summer School of Surveying, Columbia University, New York

HENRY J. BURT, B. S., C. E.

General Manager for Holabird and Roche, Architects
Member, American Society of Civil Engineers
Member, Western Society of Civil Engineers
Member, Society for the Promotion of Engineering Education

RICHARD I. D. ASHBRIDGE

Civil Engineer
Member, American Society of Civil Engineers

HERMAN K. HIGGINS

Civil Engineer
Associate Member, American Society of Civil Engineers
Member, Boston Society of Civil Engineers
Member, New England Water Works Association
Member, American Railway Bridge and Building Association

ALFRED E. PHILLIPS, C. E., Ph. D.

Professor of Civil Engineering, Armour Institute of Technology

H. E. MURDOCK, M. E., C. E.

Head of Department of Agricultural Engineering, Montana State College, Bozeman, Montana
Formerly Irrigation Engineer, U. S. Department of Agriculture

A. B. McDANIEL, B. S.

Formerly Assistant Professor of Civil Engineering, University of Illinois
Member, American Society of Civil Engineers
Member, Society for the Promotion of Engineering Education
Fellow, Association for the Advancement of Science
Author of "Excavating Machinery"

GLENN M. HOBBS, Ph. D.

Secretary and Educational Director, American School of Correspondence
Formerly Instructor, Department of Physics, University of Chicago
American Physical Society

THOMAS FLEMING, Jr., B. S., C. E.

With Chester & Fleming, Hydraulic and Sanitary Engineers
Associate Member, American Society of Civil Engineers
Member, New England Water Works Association
Member, Engineers' Society of Pennsylvania

CHARLES E. MORRISON, C. E., Ph. D.

Formerly Instructor in Civil Engineering, Columbia University
Associate Member, American Society of Civil Engineers
Author of "Highway Engineering," "High Masonry Dam Design"

EDWARD B. WAITE

Formerly Dean, and Head, Consulting Department, American School of Correspondence
American Society of Mechanical Engineers
Boston Society of Civil Engineers

C. A. MILLER, Jr.

Associate Editor, American Technical Society
Affiliated Member, Western Society of Engineers
Member, American Association of Engineers
Member, Illinois Society of Architects

JESSIE M. SHEPHERD, A. B.

Head, Publication Department, American Technical Society

Authorities Consulted

THE editors have freely consulted the standard technical literature of America and Europe in the preparation of these volumes. They desire to express their indebtedness, particularly, to the following eminent authorities, whose well-known treatises should be in the library of everyone interested in Civil Engineering.

Grateful acknowledgment is here made also for the invaluable co-operation of the foremost Civil, Structural, Railroad, Hydraulic, and Sanitary Engineers and Manufacturers in making these volumes thoroughly representative of the very best and latest practice in every branch of the broad field of Civil Engineering.

WILLIAM G. RAYMOND, C. E.

Dean of the School of Applied Science and Professor of Civil Engineering in the State University of Iowa; American Society of Civil Engineers
Author of "A Textbook of Plane Surveying," "The Elements of Railroad Engineering"

JOSEPH P. FRIZELL

Hydraulic Engineer and Water-Power Expert; American Society of Civil Engineers
Author of "Water Power, the Development and Application of the Energy of Flowing Water"

FREDERICK E. TURNEAURE, C. E., Dr. Eng.

Dean of the College of Engineering and Professor of Engineering, University of Wisconsin
Joint Author of "Public Water Supplies," "Theory and Practice of Modern Framed Structures," "Principles of Reinforced Concrete Construction"

HENRY N. OGDEN, C. E.

Professor of Sanitary Engineering, Cornell University
Author of "Sewer Design"

DANIEL CARHART, C. E.

Emeritus Professor of Civil Engineering, University of Pittsburgh
Author of "Treatise on Plane Surveying"

HALBERT P. GILLETTE

Editor of *Engineering and Contracting;* American Society of Civil Engineers; Formerly Chief Engineer, Washington State Railroad Commission
Author of "Handbook of Cost Data for Contractors and Engineers"

CHARLES E. GREENE, A. M., C. E.

Late Professor of Civil Engineering, University of Michigan

A. PRESCOTT FOLWELL

Editor of *Municipal Journal and Engineer;* Formerly Professor of Municipal Engineering, Lafayette College
Author of "Water Supply Engineering," "Sewerage"

IRVING P. CHURCH, C. E.

Professor of Applied Mechanics and Hydraulics, Cornell University
Author of "Mechanics of Engineering"

PAUL C. NUGENT, A. M., C. E.

Professor of Civil Engineering, Syracuse University
Author of "Plane Surveying"

FRANK W. SKINNER, C. E.

Consulting Engineer; Associate Editor of *The Engineering Record*
Author of "Types and Details of Bridge Construction"

HANBURY BROWN, K. C. M. G.

Member of the Institution of Civil Engineers
Author of "Irrigation, Its Principles and Practice"

SANFORD E. THOMPSON, S. B., C. E.

American Society of Civil Engineers
Joint Author of "A Treatise on Concrete, Plain and Reinforced"

JOSEPH KENDALL FREITAG, B. S., C. E.

American Society of Civil Engineers
Author of "Architectural Engineering," "Fireproofing of Steel Buildings," "Fire Prevention and Fire Protection"

AUSTIN T. BYRNE, C. E.

Civil Engineer
Author of "Highway Construction," "Inspection of Materials and Workmanship Employed in Construction"

JOHN F. HAYFORD, C. E.

Expert Computer and Geodesist, U. S. Coast and Geodetic Survey
Author of "A Textbook of Geodetic Astronomy"

WALTER LORING WEBB, C. E.

Consulting Civil Engineer; American Society of Civil Engineers
Author of "Railroad Construction in Theory and Practice," "Economics of Railroa[d] Construction," etc.

EDWARD R. MAURER, B. C. E.

Professor of Mechanics, University of Wisconsin
Joint Author of "Principles of Reinforced Concrete Construction"

HERBERT M. WILSON, C. E.

Geographer and Former Irrigation Engineer, United States Geological Survey; American Society of Civil Engineers
Author of "Topographic Surveying," "Irrigation Engineering," etc.

MANSFIELD MERRIMAN, C. E., Ph. D.

Consulting Engineer
Formerly Professer of Civil Engineering, Lehigh University
Author of "The Elements of Precise Surveying and Geodesy," "A Treatise on Hydraulics," "Mechanics of Materials," "Retaining Walls and Masonry Dams," "Introduction to Geodetic Surveying," "A Textbook on Roofs and Bridges," "A Handbook for Surveyors," "American Civil Engineers' Pocket Book"

DAVID M. STAUFFER

American Society of Civil Engineers; Institution of Civil Engineers; Vice-President, Engineering News Publishing Co.
Author of "Modern Tunnel Practice"

CHARLES L. CRANDALL

Professor of Railroad Engineering and Geodesy in Cornell University
Author of "A Textbook on Geodesy and Least Squares"

N. CLIFFORD RICKER, M. Arch.

Professor of Architecture, University of Illinois; Fellow of the American Institute of Architects and of the Western Association of Architects
Author of "Elementary Graphic Statics and the Construction of Trussed Roofs"

W. H. SEARLES, C. E.

Author of "Field Engineering" and "Railroad Spiral"

HENRY T. BOVEY

Late Rector of Imperial College of Science and Technology, London, England
Author of "Treatise on Hydraulics"

WILLIAM H. BIRKMIRE, C. E.

Author of "Planning and Construction of High Office Buildings," "Architectural Iron and Steel, and Its Application in the Construction of Buildings," "Compound Riveted Girders," "Skeleton Structures," etc.

IRA O. BAKER, C. E.

Professor of Civil Engineering, University of Illinois
Author of "A Treatise on Masonry Construction," "Engineers' Surveying Instruments, Their Construction, Adjustment, and Use," "Roads and Pavements"

JOHN CLAYTON TRACY, C. E.

Assistant Professor of Structural Engineering, Sheffield Scientific School, Yale University
Author of "Plane Surveying: A Textbook and Pocket Manual"

FREDERICK W. TAYLOR, M. E.

Joint Author of "A Treatise on Concrete, Plain and Reinforced"

J. B. JOHNSON, C. E.

Author of "Materials of Construction;" Joint Author of "Design of Modern Frame Structures"

FRANK E. KIDDER, C. E., Ph. D.

Consulting Architect and Structural Engineer; Fellow of the American Institute of Architects
Author of "Architect's and Builder's Pocketbook," "Building Construction and Superintendence, Part I, Masons' Work; Part II, Carpenters' Work; Part III, Trussed Roofs and Roof Trusses," "Strength of Beams, Floors, and Roofs"

WILLIAM H. BURR, C. E.

Professor of Civil Engineering, Columbia University; Consulting Engineer; American Society of Civil Engineers; Institution of Civil Engineers
Author of "Elasticity and Resistance of the Materials of Engineering;" Joint Author of "The Design and Construction of Metallic Bridges," "Suspension Bridges, Arch Ribs, and Cantilevers"

WILLIAM M. GILLESPIE, LL. D.

Formerly Professor of Civil Engineering in Union University
Author of "Land Surveying and Direct Leveling," "Higher Surveying"

GEORGE W. TILLSON, C. E.

Past President of the Brooklyn Engineers' Club; American Society of Civil Engineers; American Society of Municipal Improvements
Author of "Street Pavements and Street Paving Material"

CHARLES E. FOWLER

Consulting Civil Engineer; Member, American Society of Civil Engineers
Author of "Practical Treatise on Subaqueous Foundations"

W. M. PATTON

Late Professor of Engineering at the Virginia Military Institute
Author of "A Treatise on Civil Engineering"

CONCRETE PAVEMENT JOINED TO TIMBER MATTRESS ON RED RIVER LEVEES

Woven wire reinforcement consists of 6- by 6-inch mesh, No. 10 gage wires in each direction. The completed concrete surface, over which is laid a layer of riprap in order to complete the levee surface, is shown in the background.

Courtesy of Clinton Wire Cloth Company, Clinton, Massachusetts

Foreword

OF all the works of man in the various branches of engineering, none are so wonderful, so majestic, so awe-inspiring as the works of the Civil Engineer. It is the Civil Engineer who throws a great bridge across the yawning chasm which seemingly forms an impassable obstacle to further progress. He designs and builds the skeletons of steel to dizzy heights, for the architect to cover and adorn. He burrows through a great mountain and reaches the other side within a fraction of an inch of the spot located by the original survey. He scales mountain peaks, or traverses dry river beds, surveying and plotting hitherto unknown, or at least unsurveyed, regions. He builds our Panama Canals, our Arrow Rock and Roosevelt Dams, our water-works, filtration plants, and practically all of our great public works.

¶ The importance of all of these immense engineering projects and the need for a clear, non-technical presentation of the theoretical and practical developments of the broad field of Civil Engineering has led the publishers to compile this great reference work. It has been their aim to fulfill the demands of the trained engineer for authoritative material which will solve the problems in his own and allied lines in Civil Engineering, as well as to satisfy the desires of the self-taught practical man who attempts to keep up with modern engineering developments.

¶ Books on the several divisions of Civil Engineering are many and valuable, but their information is too voluminous to be of the greatest value for ready reference. The Cyclopedia of Civil Engineering offers more condensed and less technical treatments of these same subjects from which all unnecessary duplication has been eliminated; when compiled into nine handy volumes, with comprehensive indexes to facilitate the looking up of various topics, they represent a library admirably adapted to the requirements of either the technical or the practical reader.

¶ The Cyclopedia of Civil Engineering has for years occupied an enviable place in the field of technical literature as a standard reference work and the publishers have spared no expense to make this latest edition even more comprehensive and instructive.

¶ In conclusion, grateful acknowledgment is due to the staff of authors and collaborators—engineers of wide practical experience, and teachers of well recognized ability — without whose hearty co-operation this work would have been impossible.

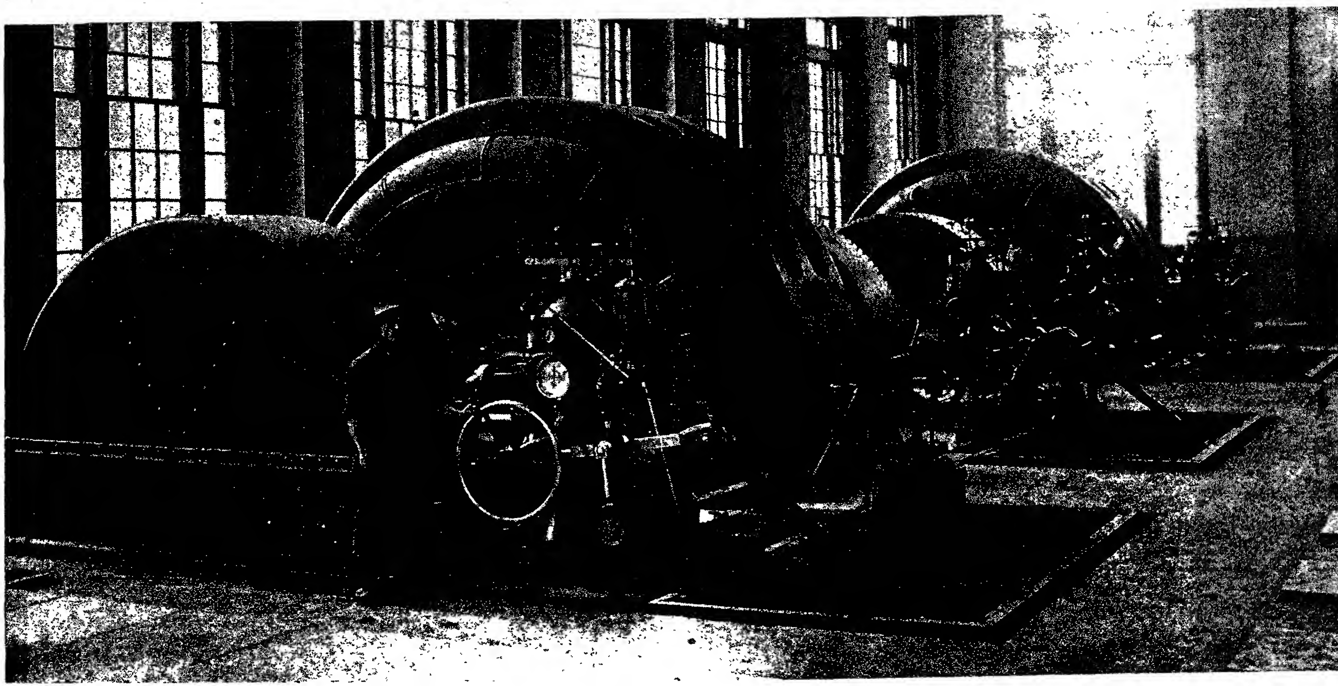
Table of Contents

VOLUME VIII

---

*For page numbers, see foot of pages.

†For professional standing of authors, see list of Authors and Collaborators at front of volume.



GENERATING UNITS OF BIG CREEK DEVELOPMENT OF THE PACIFIC LIGHT AND POWER CORPORATION

The main units are of the two-runner type, the wheel disks being pressed onto the projecting overhung ends of the generator shaft. The turbines are the largest impulse units ever constructed and operate under one of the highest heads in the world. Capacity of each generator, 11,000 K. W.

Stone & Webster Construction Company, Construction Engineers

WATER-POWER DEVELOPMENT

PART I

THEORY OF WATER-POWER

1. Introduction. One of the fundamental teachings of science is that all energy in the solar system is derived from the sun. Through the agency of that luminary, water from the earth's oceans, seas, and lakes is transformed into vapor, and in this condition is diffused throughout the atmosphere, transported by the winds—themselves created by this same solar energy—over long distances and wide areas, and finally precipitated over land and water, hills and valleys, mainly in the form of rain and snow. Of the total precipitation on the continents, part is evaporated from land and water surfaces, vegetation, etc.; part runs off more or less rapidly as surface flow into the nearby drainage channels, and thence, more or less directly, to the ocean; and part sinks into the ground. Of this last, a portion is retained by capillary attraction within reach of vegetation, to be taken up slowly by the rootlets and transpired through the leaves; the balance percolates downward until it reaches the surface of the underground water flow, which it joins in its relatively slow motion to some nearby stream, lake, or other drainage course, or directly to the ocean. It is then again evaporated into the atmosphere, with a continuous repetition of the cycle described above.

Thus every elevated body of water, every running stream, is a source of power whose energy has been derived or borrowed from the sun; and under proper conditions, a large proportion of this energy may be transformed into useful work.

DEFINITIONS OF UNITS AND TERMS

2. Unit of Work. The unit of work in general use is the *foot-pound* (ft.-lb.), representing the work of lifting a mass of one pound a height of one foot against gravity. Since the force of gravity, and therefore the weight of a given mass, is not constant for all points on the surface of the earth, it follows that the foot-pound, or *gravita-*

tion-measure of work, is not a constant unit. Its variation, however, is so small as to be negligible for ordinary purposes; and, being much simpler than the theoretically accurate units which must occasionally be employed in scientific investigation, it has remained in very general use. Thus the work done in raising 20 pounds of water through a height of 1 foot, or 1 pound of water through a height of 20 feet, or 5 pounds of water through a height of 4 feet, or in fact any other combination of pounds of water and feet of height making a product of 20, is said to be 20 foot-pounds.

3. Power. In the preceding definition, the element of *time* was not considered; thus, in the above example, 20 foot-pounds of work were done, whether the indicated operation took one minute to perform or extended over a period of one hour, or longer. The term *power* is defined as the amount of energy that can be exerted, or work done, *in a given time.*

Unit of Power. For industrial purposes, the unit most commonly employed is the *horse-power* (h.p.), which represents the capacity to perform 33,000 foot-pounds of work in one minute, or 550 foot-pounds of work in one second; it thus indicates the *rate of work.*

Example. A pump raising 7,500,000 gallons of water* in 10 hours to an elevated tank 50 feet high, is performing:

$$\frac{7{,}500{,}000 \times 62.5 \times 50}{7.5} = 3{,}125{,}000{,}000 \text{ ft.-lbs. of useful work; or,}$$

$$\frac{3{,}125{,}000{,}000}{10 \times 60} = 5{,}208{,}333 \text{ ft.-lbs. per minute,}$$

which is equivalent to:

$$\frac{5{,}208{,}333}{33{,}000} = 157.8 \text{ h.p.}$$

This amount of horse-power is the rate of work which, in the example above, must be continued for 10 hours in order to raise the total quantity of water. The entire problem may be conveniently performed in one operation, thus:

$$\frac{7{,}500{,}000 \times 62.5 \times 50}{7.5 \times 10 \times 60 \times 33{,}000} = 157.8 \text{ h.p.}$$

4. Energy. The amount of energy existing in any agent is measured by the quantity of work it is able to do; *energy* and *work*

*One cubic foot of water weighs 62.5 lbs. and contains 7.5 gallons (approximately).

are therefore measured by the same unit. "When energy is exerted, work is done against resistance." As usually stated in Theoretical Mechanics, energy may exist as *potential energy*—energy of position; or *kinetic energy*—energy of motion; or partly in one form, and partly in the other. Thus (see Fig. 1) a cannon-ball weighing W pounds, located in an elevated position h feet above any plane of reference, possesses Wh foot-pounds of potential energy with respect to that plane, by virtue of its position. If it be allowed to fall to the plane, it will, at its lowest point, theoretically have acquired a velocity of $v(=\sqrt{2gh})$ feet per second, and will therefore, at that level, possess kinetic energy to the amount of $W\frac{v^2}{2g}(=Wh)$ foot-pounds by reason of its motion. Further, if we analyze the conditions at some intermediate plane h_1 feet below its original position, and h_2 feet above the lower level, we shall find that the ball has acquired at this point a velocity of v_1 $(=\sqrt{2gh_1})$ feet per second, and therefore possesses kinetic energy to the amount of $W\frac{v_1^2}{2g}$ $(=Wh_1)$ foot-pounds due to its motion; but, by reason of its position h_2 feet above the lower plane, it still possesses Wh_2 foot-pounds of potential energy; consequently, with respect to the lower plane, the ball possesses a total energy represented by $W\left(\frac{v_1^2}{2g}+h_2\right)=W(h_1+h_2)=Wh=W\frac{v^2}{2g}$ foot-pounds.

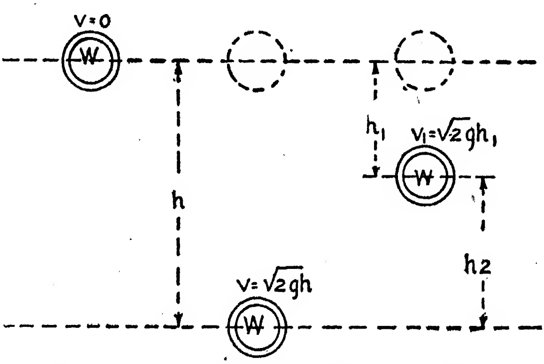


Fig. 1. Illustrating Relation between Potential and Kinetic Energy.

Thus potential and kinetic energies are mutually convertible, theoretically without loss; practically, more or less energy will be transformed into heat during the conversion, and dissipated. But the great principle of the Conservation of Energy teaches that the *total quantity* of energy existing, or stored in the ball in any position, is theoretically a constant quantity.

5. Pressure-, Velocity-, and Gravity-Head. In hydraulic work, because of the nature of the medium dealt with—water being considered in this connection a perfect fluid, and incompressible—and because of the character of the problems presented, it is customary and convenient to consider the energy of water as capable of existing in three forms—*Pressure*, *Velocity*, and *Gravity*. Thus, in Fig. 2, with the conditions as represented (see also "Hydraulics," page 34), if the valve at D be closed, the water will rise in tube CC (called a *piezometer tube*) to the same level EF as that existing in the reservoir, and the pressure in the pipe at C will be represented by the head h

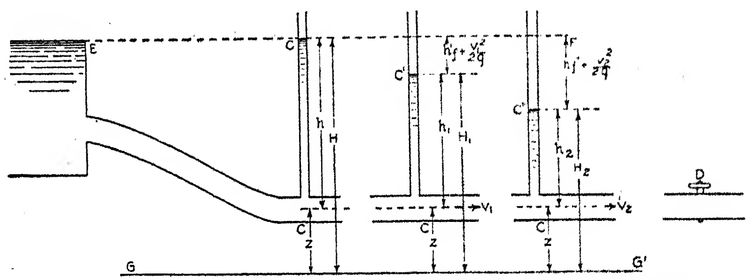


Fig. 2. Illustrating Relations of Pressure-, Velocity-, and Gravity-Head.

feet. Now, if the valve at D be partially opened, so that there is some velocity of flow v_1, in the pipe at section C, the column of water in the tube CC will sink to some lower level, as CC', and the pressure in the pipe at C will be that due to the head h_1 feet. Similarly, if the valve be now completely opened, so that the velocity of flow v_2, in the same section, becomes greater than v_1, the column of water in the tube will sink still lower, as CC'', indicating a pressure in the pipe at C represented by the head h_2 feet. If the loss of head in friction, etc., in the two cases of flow indicated above be respectively represented by h_f' and h_f'', the important relations existing are clearly shown in this diagram. It is evident that at the end of the pipe, where the water discharges freely into the air, no pressure-head exists, all the energy possessed by the issuing water being kinetic.

6. Total Head. Now let GG' represent any horizontal plane of reference—for example, the level of the tail-race water in a hydraulic power plant. With reference to this plane, the total effective head existing in the pipe at the section C, is:

(*a*) *For the case of no flow—*

$$z + h = H \text{ feet};$$

(*b*) *For the case of partial flow—*

$$z + h_1 + \frac{v_1^2}{2g} = H - h'_f \text{ feet};$$

(*c*) *For the case of full flow—*

$$z + h_2 + \frac{v_2^2}{2g} = H - h''_f \text{ feet}.$$

The distance z may be called the *gravity-head* (it corresponds to the head in potential energy already referred to); $\frac{v_1^2}{2g}$ and $\frac{v_2^2}{2g}$ are properly termed the *velocity-heads* (they correspond to the heads in kinetic energy already explained); h_1 and h_2 are known as the *pressure-heads* (see "Hydraulics," Article 6); h'_f and h''_f represent the heads lost in overcoming the various resistances to flow, principally friction in the pipe for the usual cases; but in the general case they include losses of head due to entrance, valves, curves, etc. (see "Hydraulics," Articles 28 and 34).

7. Energy per Pound of Water. The quantities stated above as number of feet in (*a*), (*b*), and (*c*) may be understood in another sense. Each may represent the total number of foot-pounds of energy existing *in every pound of water* in, or passing through, the pipe at section C; thus,

(*a*)	$z + h =$	H	foot-pounds per pound of water
(*b*)	$z + h_1 + \frac{v_1^2}{2g}$	$= H - h'_f$	" " " " " "
(*c*)	$z + h_2 + \frac{v_2^2}{2g}$	$= H - h''_f$	" " " " " "

8. Total Energy. Now suppose W_1 and W_2 pounds of water per second respectively to pass the section C in the two cases of flow considered; then, with respect to the plane GG', the total energy of the water as it passes this section is, for the one case:

$$(b) \quad W_1 \left(z + h_1 + \frac{v_1^2}{2g}\right) \text{ foot-pounds};$$

and for the other:

$$(c) \quad W_2 \left(z + h_2 + \frac{v_2^2}{2g}\right) \text{ foot-pounds};$$

and these expressions represent, for the two cases considered, the *total amount of energy* possessed by the water, *with respect to the plane GG′*, and theoretically capable of being delivered to a machine

or motor, by the descent of the water from the upper level EF to the lower level GG'.

Where the water issues freely into the air from the extremity of the pipe, or through a nozzle at the end, no pressure exists; therefore, in the expression corresponding to (*b*) or (*c*), above, for such section, the term representing pressure-head disappears, leaving the two terms indicating gravity-head and velocity-head.

Further, if the plane of reference passes through the center of the end of the pipe or nozzle opening, the term representing gravity-head also disappears, leaving the velocity-head alone to indicate the energy of the stream at this point.

It is usually more convenient to express the sum of gravity-head and pressure-head in a single term: thus, $z + h_1 = H_1$; and $z + h_2 = H_2$; here H_1 and H_2 may be called the *piezometer heights*.

9. Efficiency. The efficiency of an apparatus for utilizing the kinetic energy of moving water, or the potential energy of still water, is *the ratio of the amount of work given out by the apparatus to the amount of work delivered to it*; or, as it is sometimes stated, it is *the ratio of the useful work to the theoretic energy.* This topic will be treated more fully in a later article; for the present, if e represent the efficiency of a motor, then,

$$e = \frac{\text{Foot-pounds or horse-power } \textit{given out by} \text{ motor}}{\text{Foot-pounds or horse-power } \textit{delivered to} \text{ motor}}.$$

As will be seen later, the denominator does not represent the full theoretic energy of the waterfall, since more or less of this energy must be utilized in overcoming the resistances encountered in conducting the water to the motor.

Examples. 1. A motor is operated by a stream of water discharged through a 2-foot pipe with a velocity of 10 feet per second. The motor gives out at its shaft 4.4 horse-power. What is the efficiency of the motor?

$$\frac{3.1416 \times 10 \times 62.5 \times 100}{550 \times 64.4} = 5.5 \text{ horse-power delivered to motor}$$

$$e = \frac{4.4}{5.5} = 80 \text{ per cent efficiency.}$$

2. A small turbine wheel using 100 cubic feet of water per minute under a head of 45 feet, is found to give 6 horse-power. What is the efficiency of the wheel?

$$6 \text{ Horse-Power} = 6 \times 33{,}000 = 198{,}000 \text{ ft.-lbs. per min.}$$

$$e = \frac{198{,}000}{100 \times 62.5 \times 45} = 70.4 \text{ per cent efficiency.}$$

10. Theoretic Efficiency. If the efficiency of the motor actuated by the water were 100 per cent, it would give out at its shaft, as useful work, the same number of foot-pounds that were delivered to it. It is also interesting to note that if the efficiency of the hydraulic parts of the plant were 100 per cent—that is, if there were no hydraulic losses of head—the total energy of the water (see Fig. 2) represented by the total head H feet, or H foot-pounds per pound of water, would be available; and, if operating a motor of 100 per cent efficiency, the total energy of the water would be given out as useful work at the shaft of the motor. In practice these ideal conditions can never be fully realized, for there are certain hydraulic and mechanical losses of energy, which, while they may be reduced to the lowest limits by means of proper design, nevertheless, cannot be entirely eliminated.

Examples. 1. A pond containing 2,000,000 cubic feet of water is at an average elevation of 50 feet above the lower level. How much potential energy does this theoretically represent at the lower level?

$$2{,}000{,}000 \times 62.5 \times 50 = 6{,}250{,}000{,}000 \text{ ft.-lbs.}$$

If this water is fed to a small motor at the rate of 100 cubic feet per minute, what horse-power does this represent, and how long may the motor be operated?

$$\frac{100 \times 62.5 \times 50}{33{,}000} = 9.5 \text{ h.p.}$$

$$\frac{2{,}000{,}000}{100 \times 60 \times 24} = 13\tfrac{7}{8} \text{ days, or 13 days 21 hours.}$$

Assuming that the motor has an efficiency of 75 per cent, how much power may be taken off at its shaft?

$$9.5 \times .75 = 7.1 \text{ h.p.}$$

2. The discharge of a stream is 1,000 cubic feet per second; its mean velocity is 3 feet per second. What horse-power does this represent?

$$\frac{1{,}000 \times 62.5 \times (3)^2}{550 \times 64.4} = 15.88 \text{ h.p.}$$

3. Water issues from a nozzle at the rate of 50 feet per second; the area of the nozzle opening is 0.1 square foot. How many foot-pounds of kinetic energy does this represent? How many horse-power? If this jet operates a motor of 80 per cent efficiency, what horse-power will the motor actually yield?

$$0.1 \times 50 \times 62.5 \times \frac{(50)^2}{64.4} = 12{,}131 \text{ ft.-lbs. per second.}$$

$$\frac{12{,}131}{550} = 22 \text{ h.p.}$$

$$22 \times .80 = 17.6 \text{ h.p.}$$

FLOW THROUGH NOZZLES

11. *Pipe End with Nozzle. *Pressure at Base of Nozzle.* For many purposes water is delivered at considerable velocity through a nozzle at the end of a pipe. It is therefore desirable to develop a formula for velocity of flow, and quantity of discharge, for such cases.

If the pressure-head h_1 (Fig. 3) at the entrance or base of a *smooth* nozzle be observed, either by a piezometer tube or by a pressure

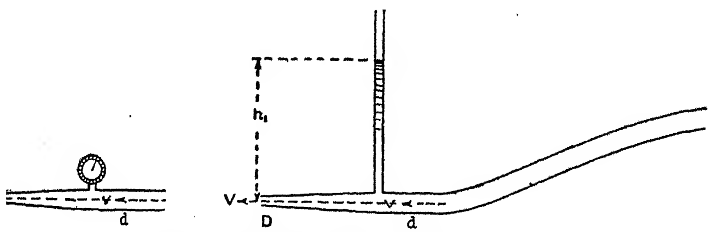


Fig. 3. Pipe with Nozzle Attachment.

gauge, then, since the nozzle velocity V is a consequence of the pressure-head h_1 and the velocity-head $\frac{v^2}{2g}$ of the water in the pipe approaching the nozzle with a velocity of v feet per second, the real or effective head on the nozzle is $h_1 + \frac{v^2}{2g}$; the theoretic velocity from the nozzle is:

$$V = \sqrt{2g\left(h_1 + \frac{v^2}{2g}\right)};$$

and the actual velocity is:

$$V = c_1\sqrt{2g\left(h_1 + \frac{v^2}{2g}\right)},$$

in which c_1 denotes the coefficient of velocity, which, for smooth nozzles, is the same as the coefficient of discharge. In these equations, h_1 is expressed in feet; V and v in feet per second. Let D and d be the diameters, in feet, of the nozzle and pipe respectively.

Since the Discharge q = Area $\times$ Velocity,

$$q = \frac{\pi D^2}{4}V = \frac{\pi d^2}{4}v;$$

*The analytical treatment throughout this work follows closely that of Professor Mansfield Merriman, "Treatise on Hydraulics," John Wiley & Sons, New York.

$$v = (\frac{D}{d})^2 V.$$

Substituting this value of v in the equation above, and solving for V, there results:

$$V = \sqrt{\frac{2gh_1}{(\frac{1}{c_1})^2 - (\frac{D}{d})^4}} \quad \ldots\ldots\ldots\ldots(1)$$

in feet per second; and the discharge (area × V) is:

$$q = 0.7854\ D^2 \sqrt{\frac{2gh_1}{(\frac{1}{c_1})^2 - (\frac{D}{d})^4}} \quad \ldots\ldots\ldots(2)$$

in cubic feet per second; and the velocity-head of the issuing jet is:

$$\frac{V^2}{2g} = \frac{h_1}{(\frac{1}{c_1})^2 - (\frac{D}{d})^4} \quad \ldots\ldots\ldots\ldots(3)$$

In many cases it is common to read the pressure at the base of the nozzle in pounds per square inch and the diameters of nozzle and pipe in inches; then h_1 (in feet) equals $2.304p_1$ (in pounds per square inch), while d_1 and D_1 in inches equal $\frac{d}{12}$ and $\frac{D}{12}$, respectively; and the discharge is frequently stated in gallons per minute. Making substitutions in Equation 2, we have (gallons per minute):

$$q = 29.83D_1^2\sqrt{\frac{p_1}{(\frac{1}{c_1})^2 - (\frac{D_1}{d_1})^4}} \quad \ldots\ldots\ldots(4)$$

Example: The pressure-gauge at the base of a smooth $1\frac{1}{4}$-inch nozzle reads 80 pounds per square inch; compute the velocity and discharge from the nozzle, the velocity-head of the issuing stream, and the mean velocity in the pipe, if the latter be $2\frac{1}{2}$ inches in diameter. Assume 0.97 as the value of the coefficient.

Substituting given numerical values in the equations, we have:

$$V = \sqrt{\frac{64.4 \times (2.304 \times 80)}{(\frac{1}{.97})^2 - (\frac{1}{2})^4}} = 108.9 \text{ feet per second}$$

$$q = \text{Area} \times V = \frac{0.7854 \times 1.25^2}{144} \times 108.9 = 0.93 \text{ cubic foot per second}$$

$$\frac{V^2}{2g} = \frac{(108.9)^2}{64.4} = 184.2 \text{ feet}$$

$$v = (\frac{D}{d})^2\ V = \frac{1}{4} \times 108.9 = 27.2 \text{ feet per second}$$

What horse-power does this represent?

$$\frac{0.93 \times 62.5}{550} \times 184.2 = 19.5 \text{ h.p.}$$

With a motor of 80 per cent efficiency, how much useful work will be obtained?

$$19.5 \times 0.80 = 15.6 \text{ h.p.}$$

12. Pipe Line with Nozzle. In Fig. 4, let h be the total head on the end of the nozzle, D its smaller diameter in feet, and V the velocity of the issuing stream in feet per second. Let d and v be the corresponding quantities for the pipe or hose, and l its length in feet. Of

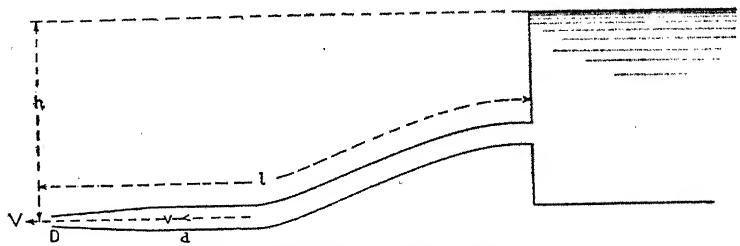


Fig. 4. Loss of Head in Pipe and Nozzle.

the total available head h on the end of the nozzle, only $\frac{V^2}{2g}$ remains; so that $h - \frac{V^2}{2g}$ represents the head lost or dissipated in overcoming various resistances to flow, from the reservoir to the tip of the nozzle. This lost head consists of several parts (see "Hydraulics," Article 34); we may therefore write:

$$h - \frac{V^2}{2g} = \left\{ \left(\frac{1}{c}\right)^2 - 1 \right\} \frac{v^2}{2g} + \frac{8gl}{C^2 d}\,\frac{v^2}{2g} + m\,\frac{v^2}{2g} + n\,\frac{v^2}{2g} + m'\,\frac{V^2}{2g} \quad \ldots (5)$$

in which,

$\left\{ \left(\frac{1}{c}\right)^2 - 1 \right\} \frac{v^2}{2g}$ = Loss of head at entrance; $\frac{8\,gl}{C^2 d}\,\frac{v^2}{2g}$ = Head lost in friction in the pipe (see "Hydraulics," Articles 28 and 36); $m\,\frac{v^2}{2g}$ = Head lost in bends and curves; $n\,\frac{v^2}{2g}$ = Head lost by the passage of the water through valves and gates; and, lastly, $m'\frac{V^2}{2g}$ = Head lost in passing through the nozzle.

The equation for the value of m' assumes a form similar to that for entrance loss into a pipe:

$$m' = \left\{ \left(\frac{1}{c_1}\right)^2 - 1 \right\},$$

in which c_1 is the coefficient of velocity, which, for smooth nozzles, is the same as the coefficient of discharge; its value may be taken as 0.97 for such nozzles, with the small diameter between $\frac{3}{4}$ inch and $1\frac{1}{2}$ inches, under ordinary range of pressures.

Since, in steady flow, the velocities v and V are inversely proportional to the areas of the corresponding cross-sections,

$$V = v\left(\frac{d}{D}\right)^2$$

Inserting this value of V in Equation 5, and solving for v, there results:

$$v = \sqrt{\frac{2gh}{\left\{\left(\frac{1}{c}\right)^2 - 1\right\} + \frac{8gl}{C^2 d} + m + n + \left(\frac{1}{c_1}\right)^2 \left(\frac{d}{D}\right)^4}} \quad \ldots (6)$$

for the velocity of flow in the pipe, in feet per second.

The velocity and discharge from the nozzle are then:

$$V = \left(\frac{d}{D}\right)^2 v, \quad \ldots\ldots\ldots\ldots\ldots (7)$$

and,

$$q = \frac{1}{4}\pi D^2 V \quad \ldots\ldots\ldots\ldots\ldots (8)$$

In many cases the sum of the losses at entrance, through valves and gates, and around bends and curves, is sufficiently small, in comparison with the loss in pipe friction, to be negligible; in such cases, Equation 6 reduces to

$$v = \sqrt{\frac{2gh}{\frac{8gl}{C^2 d} + \left(\frac{1}{c_1}\right)^2 \left(\frac{d}{D}\right)^4}} \quad \ldots\ldots\ldots (9)$$

Example. A smooth nozzle with a small diameter of 1 inch is attached to a 3-inch pipe 1,500 feet long; the tip of the nozzle is 64 feet below the surface of the water in an elevated reservoir. Assume $C = 100$, and determine the velocity of flow in the pipe, and through the nozzle. Find also the discharge, and the efficiency of the pipe and nozzle.

Since in this case the entrance loss is relatively small, because the pipe is long in comparison with its diameter, and therefore pipe friction is relatively large, Equation 9 may be used:

$$v = \sqrt{\frac{64.4 \times 64}{\frac{8 \times 32.2 \times 1{,}500}{(100)^2 \times 0.25} + \left(\frac{1}{.97}\right)^2 \left(\frac{3}{1}\right)^4}} = 4.14 \text{ feet per second,}$$

for the velocity of flow in the pipe.

$$V = v\left(\frac{d}{D}\right)^2 = 4.14 \times 9 = 37.26 \text{ feet per second,}$$

for the velocity of the jet issuing from the nozzle.

$$q = \frac{\pi d^2}{4} v = \frac{3.1416 \times (\frac{1}{4})^2}{4} \times 4.14 = 0.20 \text{ cu. ft. per second.}$$

The energy of the jet is:

$$W \frac{V^2}{2g} = \frac{.20 \times 62.5 \times (37.26)^2}{64.4} = 269.5 \text{ ft.-lbs. per second.}$$

The theoretic energy is:

$$Wh = .20 \times 62.5 \times 64 = 800 \text{ ft.-lbs. per second.}$$

The efficiency of pipe and nozzle, therefore, is:

$$\frac{269.5}{800} = 33.7 \text{ per cent.}$$

If, under the conditions just stated, we suppose the nozzle removed, the last term in the denominator of Equation 9 will disappear, and the equation will assume the form:

$$v = \frac{C}{2} \sqrt{\frac{hd}{l}} = C \sqrt{r \frac{h}{l}} = C \sqrt{rs} \quad \ldots . \mathbf{(10)}$$

which is Equation 30 in "Hydraulics," for the case of a pipe of uniform diameter; or Equation 33, for flow in open channels.

Equation 7, taken in connection with Equation 6 or its simpler form, Equation 9, shows that the smaller the nozzle diameter compared with that of the pipe, within ordinary practical limits, the greater will be the nozzle *velocity*; but the greatest *discharge* will occur (Equation 8) when the nozzle diameter is as large as possible; that is, when it is equal to the pipe diameter—in other words, when there is no nozzle attached.

13. Relation of Pipe and Nozzle Diameters. When the object of attaching a nozzle to a pipe is to utilize the velocity-head of the issuing jet $(= \frac{V^2}{2g})$ without regard to the quantity of water discharged, a large pipe and a relatively small nozzle should be employed. When the object is to obtain as large a discharge as possible, no nozzle should be used, and the pipe should be as large as practical considerations will warrant. When the object is to utilize the energy of the jet in producing power by means of a water-motor, in which case both velocity-head and quantity of discharge are concerned, there is a definite relation existing between the diameters of nozzle and pipe that will render this a maximum.

14. Maximum Power Derivable from Nozzle Jet. From Equations 9 and 7, we derive:

$$V = \sqrt{\frac{2gh}{\frac{8gl}{C^2 d}\left(\frac{D}{d}\right)^4 + \left(\frac{1}{c_1}\right)^2}} \quad \ldots\ldots\ldots(\mathbf{11})$$

Then, if w be the weight in pounds of a cubic foot of water, we have, for the theoretical kinetic energy of the issuing jet in foot-pounds per second (weight of discharge in pounds per second $\times$ velocity-head):

$$K = w\tfrac{1}{4}\pi D^2 V \frac{V^2}{2g} = \frac{w\pi D^2 V^3}{8g} \quad \ldots\ldots(\mathbf{12})$$

Substituting in this equation the value of V above (Equation 11), and ascertaining, by the procedure usually adopted in such cases (differential calculus), the value of D to render K a maximum, we obtain:

$$D = \tfrac{1}{2}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{4}}, \quad \ldots\ldots\ldots\ldots\ldots(\mathbf{13})$$

which is a formula for diameter of nozzle in terms of diameter and length of pipe (all in feet) to produce the maximum kinetic energy of the jet issuing from the nozzle.

With a nozzle of this diameter, the velocity of the issuing jet is obtained by placing the value of D from Equation 13 in Equation 11, with the result:

$$V = 2c_1\sqrt{\frac{gh}{3}} = c_1\sqrt{2g\left(\frac{2}{3}h\right)} = 0.816c_1\sqrt{2gh} \quad \ldots(\mathbf{14})$$

Since the value of c_1 for ordinary cases is about 0.97, it may be said that the nozzle velocity necessary to produce the *maximum power* is about 80 per cent of the theoretic velocity due to the actual static head on the nozzle tip.

15. Relation between Total Head and Friction Head for Maximum Power. The relation expressed by Equation 14 leads to some interesting conclusions. Since $V = .80\sqrt{2gh}$ for maximum power, $\frac{V^2}{2g} = .64\,h$; therefore, since the total head is h, $.36h$ must be used in overcoming pipe and nozzle resistance, to give the most advantageous velocity for power purposes. Again, omitting nozzle resistance (as represented by c_1), $\frac{V^2}{2g} = .667h$; therefore $.333h$ must be used in overcoming pipe friction alone. That is to say, with the conditions

arranged to furnish maximum power, $\frac{1}{3}$ of the total static head on the nozzle tip is being used to overcome pipe friction, and the remaining $\frac{2}{3}h$ is transformed into the velocity-head of the issuing stream after due deduction or allowance for nozzle resistance. The second value of V (Equation 14) shows this directly. If no nozzle is attached, therefore, the conditions for maximum power obtain when $\frac{1}{3}$ the total static head is used in overcoming pipe friction, the remaining $\frac{2}{3}$ of the head being available as velocity-head, or as pressure-head, or partly in one form and partly in the other.

Usually the discharge in cubic feet per second (q) is known; then, by simple substitution (Equations 8 and 14), the values for maximum work are:

$$D = \left(\frac{12\, q^2}{\pi^2\, c_1^2\, gh}\right)^{\frac{1}{4}} \quad \ldots\ldots\ldots\ldots (\mathbf{15})$$

and, from Equations 13 and 15:

$$d = 2\left(\frac{6\, q^2\, l}{\pi^2\, C^2\, h}\right)^{\frac{1}{5}} \quad \ldots\ldots\ldots\ldots (\mathbf{16})$$

in which D and d are the diameters in feet of nozzle tip and pipe to furnish maximum power. Being stated in terms of q, l, and h, these equations are occasionally the most convenient to use in solving problems.

Example. By damming a stream, an impounding reservoir was created, capable of supplying uniformly 5.92 cubic feet of water per second to a power-house below. The nozzle tip is to be 590 feet below the average water level in the reservoir; the length of pipe is 6,000 feet from reservoir to nozzle; the pipe being of riveted steel, and making due allowance for deterioration of surface with age, C was assumed to have the low value 83. What size pipe and nozzle should be used to give the maximum power? What will be the nozzle velocity? What horse-power will be developed at the nozzle? What efficiency does this represent for pipe and nozzle? What power may be derived from a wheel of 75 per cent efficiency, driven by the jet? What is the efficiency of the whole system?

From Equation 16:

$$d = 2\left\{\frac{6 \times (5.92)^2 \times 6{,}000}{(3.1416)^2 \times (83)^2 \times 590}\right\}^{\frac{1}{5}} = 1 \text{ foot, pipe diameter.}$$

From Equation 15:

$$D = \left\{\frac{12 \times (5.92)^2}{(3.1416)^2 \times (.97)^2 \times 32.2 \times 590}\right\}^{\frac{1}{4}} = 2.67 \text{ inches, nozzle diameter,}$$

or Equation 13 may be used to determine D.

From Equation 14:

$$V = 0.816 \times 0.97 \sqrt{64.4 \times 590} = 152 \text{ feet per second, nozzle velocity.}$$

$$\text{Horse-power} = \frac{WV^2}{2g \times 550} = \frac{5.92 \times 62.5 \times (152)^2}{64.4 \times 550} = 241 \text{ h.p.}$$

$$\text{Theoretic horse-power} = \frac{Wh}{550} = \frac{5.92 \times 62.5 \times 590}{550} = 397 \text{ h.p.}$$

$$\text{Efficiency} = \frac{241}{397} = 61 \text{ per cent (nearly).}$$

$$\text{Useful work from wheel} = 241 \times .75 = 181 \text{ h.p.}$$

$$\text{Efficiency of whole system} = .75 \times .61 = 46 \text{ per cent (or } \tfrac{131}{397}\text{).}$$

16. Multiple Nozzles. Sometimes an impulse wheel is driven by means of jets issuing from two or more nozzles of the same or of different diameters. Then, for maximum power, the sum of the areas of the several nozzles must equal the area corresponding to D, as computed for a single nozzle, on the assumption that the nozzle tips are at substantially the same level, and that the coefficient c_1 has the same value for each. Thus, if there be two nozzles with diameters D_1 and D_2,

$$D_1^2 + D_2^2 = \frac{1}{4}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{2}} = \frac{C d^2}{4 c_1}\sqrt{\frac{d}{g l}} \quad \ldots\ldots (\mathbf{17})$$

One diameter, as D_1, may be assumed, and the other computed from the above relation.

If the two nozzles are of equal diameter D_1,

$$D_1^2 = \frac{1}{8}\left(\frac{C^2 d^5}{g c_1^2 l}\right)^{\frac{1}{2}};$$

therefore,

$$D_1 = \frac{1}{2}\left(\frac{C^2 d^5}{4 g c_1^2 l}\right)^{\frac{1}{4}} \quad \ldots\ldots\ldots\ldots (\mathbf{18})$$

If the value of D for one nozzle has already been determined, then, for two nozzles of equal diameter D_1, from the relation stated above,

$$\frac{2\pi D_1^2}{4} = \frac{\pi D^2}{4};$$

therefore,

$$D_1 = \frac{D}{\sqrt{2}} \quad \ldots\ldots\ldots\ldots (\mathbf{18a})$$

With three or more nozzles, of the same or of different diameters, the relation of areas stated above will furnish a means of readily determining the diameters. Thus, for three nozzles of equal diameter D_1,

$$D_1 = \frac{D}{\sqrt{3}} \quad \ldots\ldots\ldots\ldots (\mathbf{18b})$$

If the discharge q is known, an analysis similar in all respects to that above will give, in place of Equation 17:

$$D_1^2 + D_2^2 = \frac{2\,q}{\pi\,c_1}\sqrt{\frac{3}{gh}}\,; \quad \ldots\ldots\ldots (19)$$

and, in place of Equation 18:

$$D_1 = \left(\frac{3\,q^2}{\pi^2\,c_1{}^2\,gh}\right)^{\frac{1}{4}}, \quad \ldots\ldots\ldots\ldots (20)$$

which will prove more convenient for use in some problems.

Example. If, in example 9, two nozzles of equal diameter were required, the diameter of each nozzle could be determined directly from Equation 18; or more simply, from Equation 18a, since the value of D has already been found:

$$D_1 = \frac{D}{\sqrt{2}} = \frac{2.67}{1.41} = 1.9 \text{ inches for each nozzle.}$$

If three equal nozzles were required, then, from Equation 18b:

$$D_1 = \frac{D}{\sqrt{3}} = \frac{2.67}{1.73} = 1.5 \text{ inches for each nozzle.}$$

PRESSURES ON FIXED AND MOVING SURFACES

17. Impulse and Reaction of Water in Motion. Let W be the number of pounds of water discharged per second from an orifice, pipe, or nozzle, or flowing in a stream, with a uniform velocity of v feet per second; then,

$$F = W\frac{v}{g} \text{ pounds} \ldots\ldots\ldots\ldots (21)$$

is called the *impulse* of the moving water. It may be regarded as a continuous pressure in the direction of motion; and it will be exerted as such upon a surface placed in the path of the jet or stream, with an intensity varying with the conditions, and ranging to the maximum value F, above. The *reaction*, or *back-pressure*, is equal in value to the impulse, but opposite in direction. For example, if a vessel containing water be freely suspended at A (Fig. 5), and water be allowed to flow out through an orifice at B, the pressure due to the head of water h causes W pounds of water per second to be discharged with the velocity v (=theoretically $\sqrt{2gh}$) feet per second. In the direction of the jet, the impulse produces motion; in the opposite direction, it produces an equal back-pressure (action and reaction being equal in amount and opposite in direction), causing the vessel to swing to the right. The first of these forces is the *impulse*, and the

second is the *reaction* of the jet; and if a force R be applied as shown, of just sufficient intensity to prevent this motion of the vessel, its value is:

$$R = W\frac{v}{g} = F, \qquad (22)$$

which is the reaction of the jet.

The impulse or reaction of a jet issuing from an orifice is double the hydrostatic pressure on the area of the orifice. For, if a is the area of the orifice, and w the weight of a cubic unit of water, the normal hydrostatic pressure on the area of the orifice when closed (see "Hydraulics," Article 6) is:

Hydrostatic pressure $= wah$ pounds.

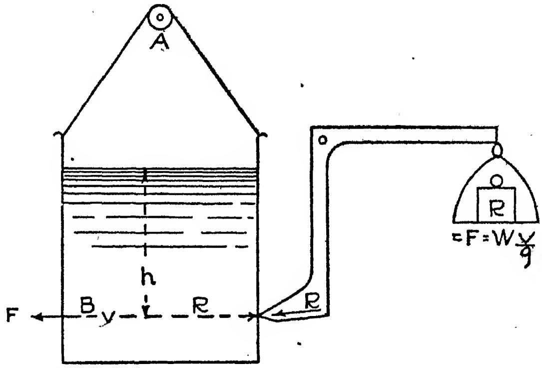


Fig. 5. Measuring the Reaction of a Jet by Weighing.

When the orifice is opened, the weight of the discharge per second (see "Hydraulics," Article 18) is theoretically $W = wav$; hence,

$$F = R = W\frac{v}{g} = wav\,\frac{v}{g} = \frac{2wav^2}{2g} = 2\,wah. \qquad (23)$$

This conclusion has been verified by many experiments (see Fig. 6).

Example. What must be the velocity of a jet of water 1 inch in diameter, issuing from a nozzle, in order that its impulse may be 100 pounds? What will be the discharge in cubic feet and in gallons per second?

$$F = \frac{Wv}{g} = \frac{wav^2}{g} = 100\ ;$$

$$\therefore v = \sqrt{\frac{100 \times 32.2}{62.5 \times .0054}} = 97.7 \text{ foot per second.}$$

$$q = av = .0054 \times 97.7 = .53 \text{ cubic foot per second.}$$

$$.53 \times 7.5 = 4 \text{ gallons per second.}$$

18. Dynamic Pressure of Water in Motion. If a jet of water strike a stationary plane normally, it produces a dynamic pressure on that plane equal to the impulse of the jet; that is:

$$P = F = W\frac{v}{g}$$

If a jet moving with a velocity v_1 be retarded by a surface so that its velocity becomes v_2, without changing its direction, the impulse in the first case is:

$$F_1 = \frac{Wv_1}{g};$$

and in the second case:

$$F_2 = \frac{Wv_2}{g};$$

and the difference,

$$P = F_1 - F_2 = W\left(\frac{v_1 - v_2}{g}\right) \quad \ldots\ldots\ldots (24)$$

is a measure of the dynamic pressure which has been developed in

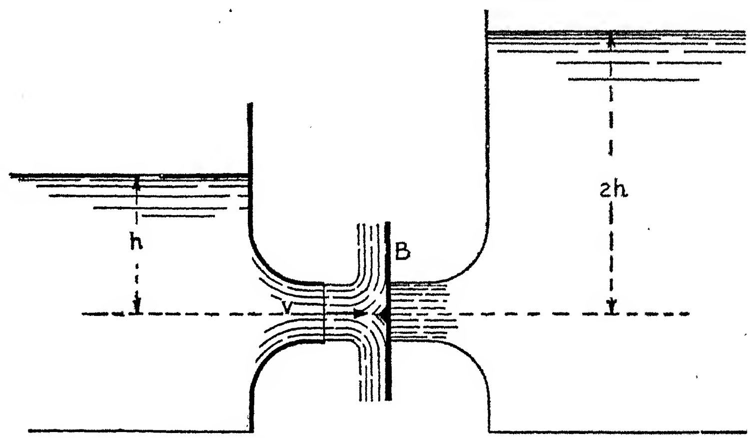


Fig. 6. Illustrating Relation between Impulse and Hydrostatic Pressure.

the direction of motion by the retardation of the velocity. If a jet of water impinge upon a stationary surface which changes its direction of motion without changing its velocity, a dynamic pressure is developed, its amount depending upon the velocity and the change in direction. In all cases this pressure is exerted upon the surface causing the retardation of velocity or change in direction of flow.

19. Static and Dynamic Pressures. *Dynamic pressure* must be clearly distinguished from *static pressure,* the laws governing in the two cases being entirely different. A static pressure due to a given head will cause a jet of water to be discharged from an orifice with a velocity proportional to the head; if this jet impinge upon a surface, a dynamic pressure will be exerted upon it, which may be equal to, greater than, or less than the static pressure due to the head,

depending upon the circumstances. Again, at any point below the surface of water, static pressure is exerted with equal intensity in all directions; dynamic pressure is exerted with different intensities in different directions.

20. Definitions. From a comparison of Equations 21 and 24, we may now define the *impulse* of a jet or stream of water as the dynamic pressure which it is capable of producing in the direction of its motion when its velocity in that direction is entirely destroyed. This may be accomplished by carefully deflecting the jet 90 degrees to its original path by means of a smooth surface, so that, no energy being dissipated in overcoming frictional or other resistances, the velocity of the water is not changed, but its component in the original direction is zero; and the *reaction* of a jet or stream of water may be defined as the backward dynamic pressure, in the line of motion, which is exerted against a vessel out of which it issues, or against a surface away from which it moves.

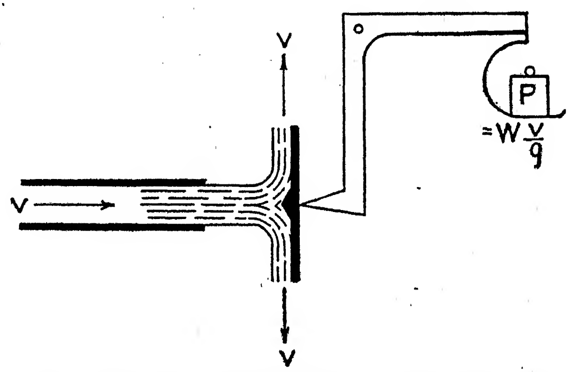


Fig. 7. Measuring Pressure of a Jet on a Plane Surface, by Weighing.

21. Laboratory Experiments on Impulse, Reaction, and Dynamic Pressure. Fig. 5 shows how the reaction of a jet may be measured; the necessary weight in the scale pan to prevent motion of the vessel has been found to be very nearly:

$$R = F = \frac{Wv}{g} = 2wa\,\frac{v^2}{2g}.$$

Fig. 6 shows how the pressure due to the impulse of a jet may be made to balance the hydrostatic pressure due to twice the head causing the flow. *B* is a loose plate with surface carefully finished to fit the mouthpiece so as to prevent leakage. Fig. 7 illustrates a simple device for measuring by weighing the dynamic pressure exerted upon a surface by the impulse of a jet impinging upon and gliding over it,

when its motion in the original direction has been entirely destroyed by being deflected 90 degrees. The result of the experiment is found to show very nearly that:

$$P = W\frac{v}{g} = 2\,wa\frac{v^2}{2g},$$

as theory requires.

Fig. 8 illustrates a case of dynamic pressure exerted upon a curved surface, due to both impulse and reaction, the former being due to the direct impact of the jet, the latter to the circumstance that the deflected stream leaves the surface in a direction which has a component of velocity parallel to the original path, but opposite in direction. Here experiment shows:

$$P > W\frac{v}{g} < 2W\frac{v}{g},$$

as theory requires.

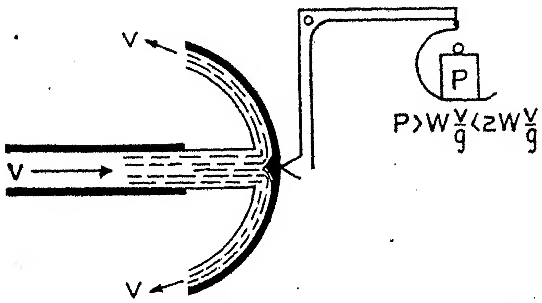


Fig. 8. Measuring Pressure from a Jet on a Curved Surface, by Weighing.

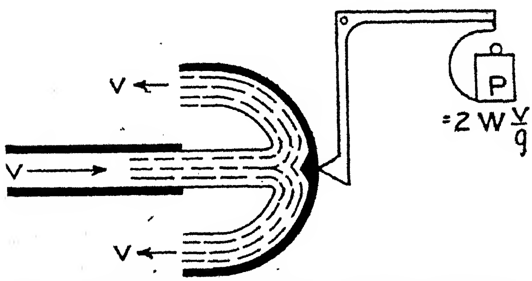


Fig. 9. Measuring Pressure from a Jet whose Direction is Completely Reversed.

Fig. 9 shows the case where the stream is deflected 180 degrees; that is, there is a complete reversal in the direction of motion; and we should expect the dynamic pressure exerted upon the surface to be equal to the sum of both impulse and reaction; namely,

$$P = F + R = 2F = 2W\frac{v}{g} = 4\,wa\frac{v^2}{2g},$$

which agrees quite closely with the results of laboratory experiments.

Example. In Fig. 7 the diameter of the tube is 1 inch; there is no contraction of the jet; and the discharge is .5 cubic foot per second. What is the velocity, and the dynamic pressure against the plane? What would be the dynamic pressure in the case represented by Fig. 9?

$$v = \frac{q}{a} = \frac{.5}{.0054} = 92.6 \text{ feet per second.}$$

$$P = W\frac{v}{g} = \frac{.5 \times 62.5 \times 92.6}{32.2} = 90 \text{ pounds.}$$

$$P = 2W\frac{v}{g} = 2 \times 90 = 180 \text{ pounds.}$$

FIXED SURFACES

22. Dynamic Pressures on Fixed Surfaces. When a stream of water impinges with a uniform velocity v on a smooth surface at rest, it glides over the surface and leaves it with the original velocity v, since there are supposed to be no frictional or other resistances, only its direction of motion being changed. The water, as it strikes the

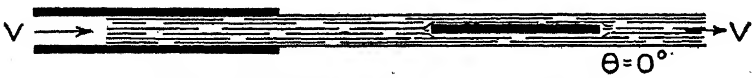


Fig. 10. Illustrating Case of No Dynamic Pressure.

surface, exerts upon it an impulse F in the direction of the path of entry; as it leaves the surface, it exerts on it an equal reaction F, in a direction opposite to its path of exit (see Figs. 11 to 14). The dynamic pressure thus developed depends upon velocity v, and change of direction of stream (angle θ). The stream is assumed to be moving horizontally while in contact with the surface, so that its velocity is not affected by gravity.

23. Resultant Dynamic Pressure. From the principle of Composition of Forces (Mechanics), the resultant dynamic pressure upon a fixed surface struck by a jet may be readily found by constructing the parallelogram of the forces of impulse and reaction, as shown in Fig. 15, in which $ab = bc = F = R$; from which we deduce (Trigonometry) that the value of this resultant pressure is:

$$P_R = F\sqrt{2(1-\cos\theta)} = 2 \text{ Sin } \frac{1}{2}\theta . W\frac{v}{g} . \quad \ldots (25)$$

and that it makes an angle of $(90° - \frac{1}{2}\theta)$ with the original direction of the jet. Its line of action passes through the intersection of F and R, and it bisects the angle between them.

24. Dynamic Pressure Parallel to Initial Direction of Jet. This is simply the component of the Resultant Dynamic Pressure in the

desired direction. From Fig. 16, this is found to be (Resolution of Forces) $ab = bc \cos(90-\frac{1}{2}\theta)$; so that,

$$P_J = P_R \cos\left(90-\frac{1}{2}\theta\right) = (1-\cos\theta)\,W\frac{v}{g} \quad . . . (26)$$

If, in this equation, $\theta = 0$, the stream glides over the surface without change of direction or retardation of velocity, and $P = 0$;

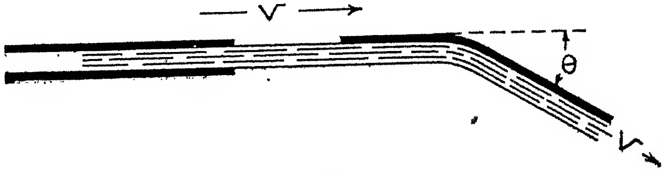


Fig. 11.

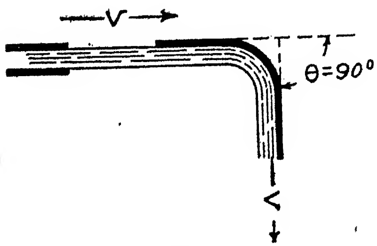


Fig. 12.

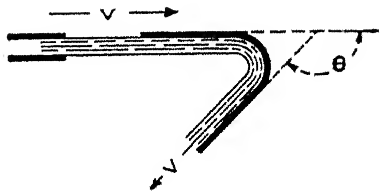


Fig. 13.

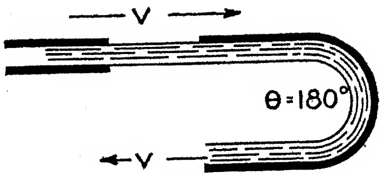


Fig. 14.

Illustrating Dynamic Pressure of Jet on Various Fixed Surfaces.

that is, no dynamic pressure is exerted upon the surface (see Fig. 10).

If $\theta = 90°$, $\cos\theta = 0$ (see Figs. 7 and 12), and therefore the dynamic pressure is:

$$P = F = W\frac{v}{g}$$

Here the escaping jet has no component of velocity normal to the surface; therefore the reaction has no influence on the pressure.

If $\theta = 180°$ (see Figs. 9 and 14), indicating a complete reversal

in the direction of the stream, $\cos \theta = -1$; hence the dynamic pressure is:

$$P = 2F = 2W \frac{v}{g}.$$

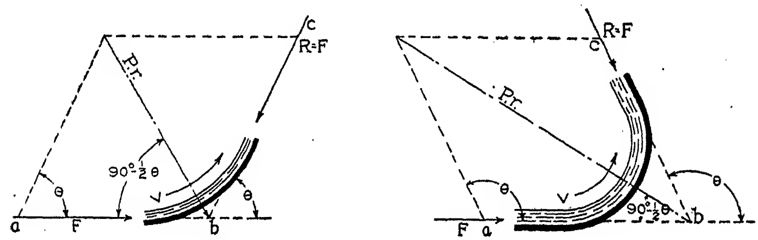


Fig. 15. Resultant Dynamic Pressure.

Here the pressure is a consequence of both impulse and reaction to their full amount.

25. Dynamic Pressure in Any Given Direction. It is frequently of importance to determine the dynamic pressure *in a given direction* exerted on a fixed surface by a stream of water. This may be ascertained by resolving the *resultant* dynamic pressure into its two components, parallel and at right angles to the required direction; the former represents the pressure in the required direction. Or the impulse and reaction may be separately resolved into their rectangular components, as above, and the algebraic sum taken of the two components parallel to the required direction. Thus, in Fig. 17, let it be required to find the dynamic pressure in a direction represented by the arrow x, which makes an angle α with the direction of the entering, and an angle θ with that of the departing stream. The components of the impulse and the reaction in the required direction, since $R = F$, are:

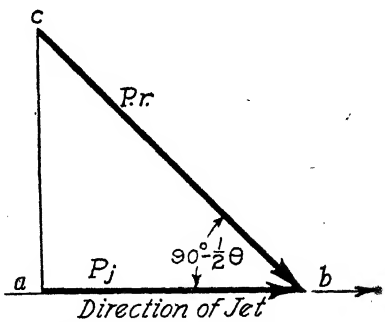


Fig. 16. Dynamic Pressure Parallel to Initial Direction of Jet.

$$P_1 = F \cos \alpha; \text{ and } P_2 = -F \cos \theta;$$

and therefore:

$$P = P_1 + P_2 = F(\cos \alpha - \cos \theta) = (\cos \alpha - \cos \theta)\, W\frac{v}{g} \quad (27)$$

If, in this general equation (27), $\alpha = 0°$,

$$P = (1 - \cos \theta) W \frac{v}{g},$$

as in Equation 26.

If $\alpha = 0°$, and $\theta = 90°$,

$$P = F = W \frac{v}{g}, \text{ as in Figs. 7 and 12}$$

If $\alpha = 0°$, and $\theta = 180°$,

$$P = 2F = 2W \frac{v}{g}, \text{ as in Figs. 9 and 14.}$$

If $\alpha = 0°$, and $\theta = 0°$, $P = 0$, as in Fig. 10.

Example: Let the jet of example in Article 11 impinge tangentially upon the fixed curved vane of Fig. 15, with $\theta = 60°$. What is the resultant dynamic pressure upon the vane, in intensity and direction? What is the dynamic pressure in a direction parallel to the jet? What is the dynamic pressure in a direction making an angle of 30 degrees with the direction of the jet?

From Equation 25 and Example in Article 11:

$$P_R = 2 \sin \tfrac{1}{2} \theta \, W \frac{v}{g}$$

$$= 2 \times \tfrac{1}{2} \times \frac{.33 \times 62.5 \times 38.5}{32.2} = 24.7 \text{ pounds.}$$

From Equation 26:

$$P_J = (1 - \cos \theta) W \frac{v}{g}$$

$$= (1 - \tfrac{1}{2}) \frac{.33 \times 62.5 \times 38.5}{32.2} = 12.4 \text{ pounds.}$$

From Equation 27 (Fig. 17):

$$P = (\cos \alpha - \cos \theta) W \frac{v}{g}$$

$$= (.866 - .500) \frac{.33 \times 62.5 \times 38.5}{32.2} = 4.5 \text{ pounds.}$$

26. Weight of Water Impinging. In all the preceding equations, W represents the weight of water in pounds per second impinging upon the surface; and, since the surface has in each case been assumed to be stationary, W is also the weight of water in pounds per second issuing from the nozzle or orifice, or flowing in the stream. It is to be clearly kept in mind that this statement is not necessarily true if the surface is supposed to move; as, for example, in the case of a jet impinging upon the vanes or blades of a water wheel. Such cases will be considered later.

27. Force and Work. It must also be clearly realized that the dynamic pressures are *forces*; they are not expressed in terms of *energy* or *work*; just as a weight resting upon a table produces *pressure* thereon, but does not perform *work*. A force must be exerted against a resistance through a definite distance, in order that work may be done; the weight may be allowed to move, and thereby compress a spring, for example, thus doing work. Similarly, the above pressures must be exerted against resistances over some definite distances, in order that work may be done. In general, if P is the

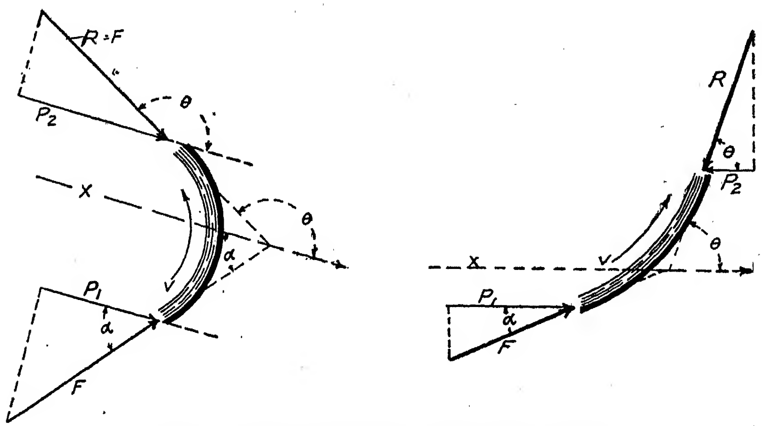


Fig. 17. Dynamic Pressure in Any Given Direction.

dynamic pressure on the surface in pounds, and if the surface is supposed to move a distance of u feet per second while overcoming some resistance, then,

$$\text{Work} = P \times u \text{ foot-pounds per second} \quad . \; . \; . \quad (28)$$

It is by reason of the dynamic pressures defined and explained above, produced by a retardation in velocity, or a change in direction of flow, that turbine wheels and other water-motors are able to transform the kinetic energy of moving water into useful work—such pressures being exerted over definite distances against resistances.

28. Losses of Energy. In the above discussion, no frictional or other losses of energy were considered. It is clear that if the surfaces are rough, or if the jet impinges on the surface in such a way as to produce "shock" or "eddies" or "foam," some of the original energy of the jet will be dissipated as heat, and the resulting pressures will be correspondingly reduced below the values indicated by the fore-

going formulæ. These losses may be largely eliminated by having the surfaces smooth and properly curved, and by so directing the jet as to strike the surface tangentially.

ABSOLUTE AND RELATIVE VELOCITIES

29. Definitions. While all velocities are in reality relative, it is convenient to define *absolute velocity* as the rate of speed of a moving object with respect to the surface of the earth; and *relative velocity* as the rate of speed of a moving object with respect to another moving body—or as the velocity the object would appear to have to a person standing upon, and viewing it from, the second moving body. In the one case, velocity is measured from, or referred to, the earth,

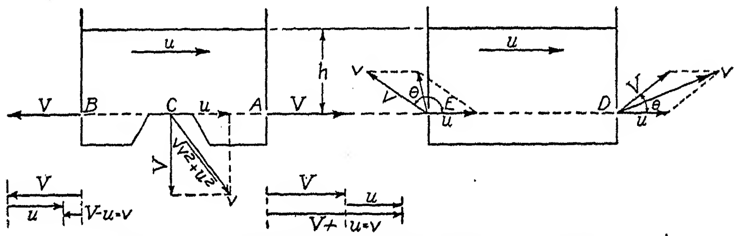


Fig. 18. Illustrating Absolute and Relative Velocities.

which is regarded as stationary; in the other case, the velocity is measured from, or referred to, the second moving body, regarded as stationary for this purpose. Thus, let Fig. 18 represent a tank so mounted that it may move horizontally to the right with a uniform absolute velocity of u feet per second; and let water issue from the various openings as indicated. Theoretically, the following absolute and relative velocities will result:

Orifice	Relative Velocity (to tank)	Absolute Velocity (referred to the earth)	θ	Cos θ
A	$V = \sqrt{2gh}$	$v = V + u$	0°	1
B	$V =$ "	$v = V - u$	180°	−1
C	$V =$ "	$v = \sqrt{V^2 + u^2}$	90°	0
D	$V =$ "	$v = \sqrt{V^2 + u^2 + 2Vu \cos \theta}$	θ	cos θ (positive)
E	$V =$ "	$v = \sqrt{V^2 + u^2 + 2Vu \cos \theta}$	θ	cos θ (negative)

The expression for absolute velocity from orifice D or E may be regarded as a general formula, and the formulæ for the other cases

may be simply derived from it by assigning the proper values to θ. These considerations of absolute and relative velocities are of great importance in determining the dynamic pressures produced by a stream of water on the moving vanes or blades of water-motors. For example, consider Fig. 19, which represents a revolving wheel having an orifice from which water issues horizontally with the relative velocity V (velocity relative to wheel), while the orifice itself is moving horizontally with an absolute velocity u (velocity relative to the ground); then, from what has preceded.

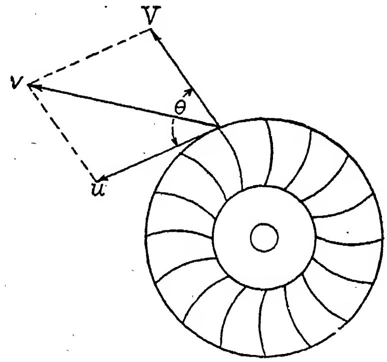


Fig. 19, Velocity of Stream Leaving or Striking Revolving Vane.

$$v = \sqrt{V^2 + u^2 + 2\,Vu\cos\theta} \quad \text{(29)}$$

is the absolute velocity of the water as it leaves the wheel (velocity with respect to the ground). In all cases, then, *the absolute velocity of a stream of water striking or leaving a moving surface is represented in magnitude and direction by the diagonal of a parallelogram of which one side is the velocity of the stream relative to the moving surface, and the other side is the absolute velocity of that surface (with reference to the ground); i. e., it is the resultant of these two velocities.*

If the directions of the component velocities lie in the same straight line, $\theta = 0°$ or $180°$; and, applying Equation 29, we derive the special formulæ:

$$v = V + u; \text{ or, } v = V - u \quad \text{(29a)}$$

MOVING SURFACES

30. Dynamic Pressure on Moving Surfaces. When a stream of water impinges upon a moving surface, the conditions are essentially different from those just discussed for surfaces at rest. Because the surface is continually moving away from the stream, two important results follow—the stream does not strike the surface with its full or absolute velocity, and the quantity of water reaching the surface per second is less than the stream discharge.

31. Case I. Jet Striking a Moving Flat Vane Normally. Let a jet (Fig. 20) whose absolute velocity is v, and cross-section a, impinge normally upon a smooth surface which is itself moving with a uniform absolute velocity u in the same direction as the jet. The *relative* velocity of the jet, or the velocity with which it strikes the surface, is $v-u$; the weight of water *leaving the orifice* per second is

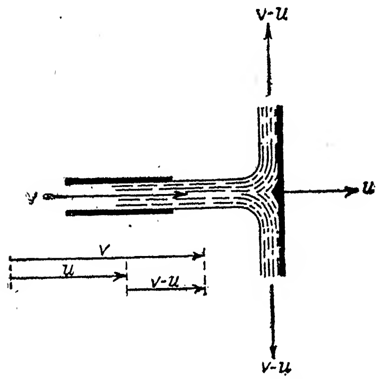


Fig. 20. Jet Striking a Moving Flat Vane Normally.

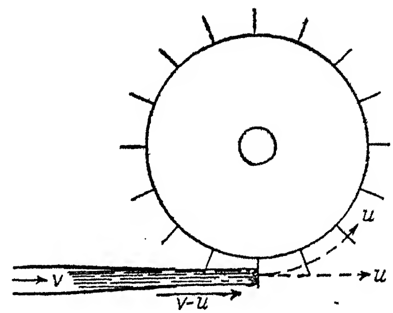


Fig. 21. Jet Striking Flat Radial Vanes of a Revolving Wheel.

$W = wav$; the weight of water *striking the surface* per second is $wa(v-u)$, if w represents the weight of a cubic unit of water; accordingly, the dynamic pressure exerted upon the surface, in the direction of motion, is:

$$P = wa\,(v-u)\,\frac{(v-u)}{g} = \frac{wa}{g}\,(v-u)^2, \quad \ldots\ldots \textbf{(30)}$$

which is equivalent to considering the surface stationary, and the stream moving with an absolute velocity of $(v-u)$ feet per second.

Work Done upon (or Given Up to) the Moving Body per Second. The work done in one second by the force P (Force $\times$ Distance) is:

$$\text{Work} = Pu = \frac{wa\,(v-u)^2\,u}{g} \quad \ldots\ldots\ldots \textbf{(31)}$$

The work is zero if $u = v$; or $u = 0$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \frac{4}{27}\,\frac{wav^3}{g} = \frac{8}{27}\,W\,\frac{v^2}{2g} \quad \ldots \textbf{(32)}$$

when $u = \frac{1}{3}\,v$.

Efficiency. Since the theoretic energy of the impinging jet is $W\frac{v^2}{2g}$, the efficiency in the case just considered is $\frac{8}{27}$, or about

30 per cent. It is evident, however, that no practical motor could be constructed on such a plan.

32. Case II. Jet Impinging on Flat Vanes of a Wheel. The resultant action of a jet impinging in rapid succession on the many flat vanes of a revolving wheel, Fig. 21, is not precisely the same as in the preceding example; but if we assume that the jet impinges normally on the vanes, and that, as the vanes come in rapid succession under the influence of the jet, and several vanes are more or less under action at the same time, the quantity of water impinging is the same as the nozzle discharge ($W = wav$); also, that the vanes move away from the jet in the direction of the latter while under impact, then we obtain for the *approximate* value of the dynamic pressure, if u represents the linear absolute velocity of the vanes at the center of impact:

$$P = W\frac{v-u}{g} = wav\frac{v-u}{g} \quad \ldots\ldots\ldots (33)$$

Work Done upon (or Given Up to) the Wheel per Second.

$$\text{Work} = Pu = \frac{wav}{g}(v-u)u \quad \ldots\ldots\ldots (34)$$

The work is zero if $u = v$, or $u = 0$; and it is a maximum and equal to:

$$\text{Work (Max.)} = \frac{1}{4}\frac{wav^3}{g} = \frac{1}{2}W\frac{v^2}{2g} \quad \ldots (35)$$

when $u = \frac{1}{2}v$.

Efficiency. Since the jet has a theoretic energy of $W\dfrac{v^2}{2g}$ foot-pounds, it is seen that the highest efficiency that can theoretically be obtained by means of a jet impinging upon rotating flat vanes is 50 per cent.

The preceding analysis applies more directly to the case of a series of flat vanes moving in a straight line, as indicated in Fig. 20, and coming in rapid succession under the influence of the jet. A motor constructed on this plan is, however, impracticable.

33. Case III. Jet Striking a Moving Curved Vane Tangentially. Fig. 22 represents a case in which the jet, with an absolute velocity v, impinges tangentially upon a vane which moves in the same direction with the uniform absolute velocity u. The velocity of the stream relative to the surface is $v-u$; and the dynamic pressure is the same as though the surface were at rest, and the stream

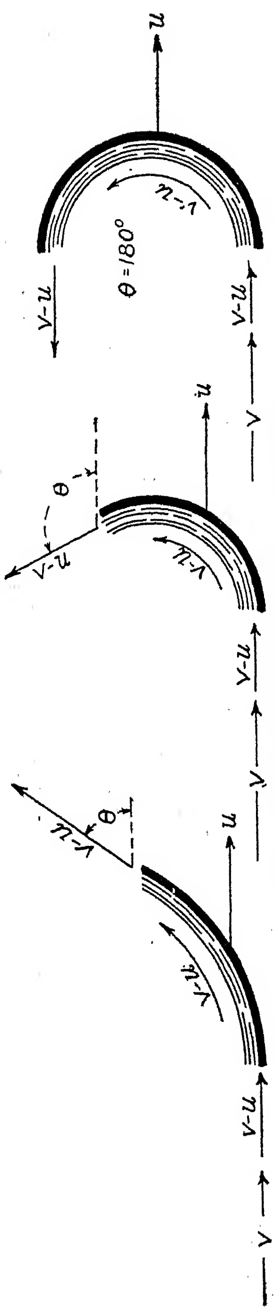


Fig. 22 Jet Striking Moving Curved Vane Tangentially.

moving and impinging with the absolute velocity $v-u$. Hence, for the dynamic pressure in the direction of the jet, we may use Equation 26, substituting $v-u$ for v; so that,

$$P = (1-\cos\theta)\, W\frac{v-u}{g} \quad \ldots\ldots (36)$$

While the dynamic pressure may be exerted with different intensities upon different parts of the vane, the total value, in the direction of motion, is that indicated by Equation 36.

Work Done. If a is the area of the cross-section of the jet, the weight of water issuing from the nozzle per second is $W = wav$; the weight striking the vane is $wa\,(v-u)$; and therefore the work is:

$$\text{Work} = Pu = (1-\cos\theta)\frac{wa}{g}(v-u)^2 u \quad \ldots\ldots\ldots\ldots\ldots\ldots (37)$$

The work is zero when $v = u$, and when $u = 0$; also when $\theta = 0°$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \frac{4}{27}(1-\cos\theta)\, wa\frac{v^3}{g} = \frac{8}{27}(1-\cos\theta)\, W\frac{v^2}{2g} \quad \ldots\ldots\ldots (38)$$

when $u = \frac{1}{3}v$.

Efficiency. Since the theoretic energy of the impinging jet is $W\frac{v^2}{2g}$, the efficiency is:

$$e = \tfrac{8}{27}(1-\cos\theta) \quad \ldots\ldots\ldots\ldots (39)$$

If $\theta = 0°$, work $= 0$, and $e = 0$; in this case the vane is a flat surface whose plane is in the direction of the stream, which therefore glides over the surface without doing work.

If $\theta = 90°$, the water leaves the vane at right angles to the direction of motion, and the maximum work, from Equation 38, is:

$$\text{Work (Max.)} = \tfrac{8}{27} W\frac{v^2}{2g} \quad \ldots\ldots\ldots (40)$$

and the efficiency is $\frac{8}{27}$, or about 30 per cent. (Compare with Equation 32.)

If $\theta = 180°$, the stream is completely reversed. In this case, (since $\cos 180° = -1$),

$$\text{Work (Max.)} = \tfrac{16}{27} W\frac{v^2}{2g} \quad \ldots\ldots\ldots (41)$$

and the efficiency is $\frac{16}{27}$, or about 60 per cent.

34. Case IV. Jet Impinging Tangentially on Curved Vanes of a Wheel. Considering a wheel with a large number of vanes, Fig. 23, and assuming the jet to impinge tangentially, and the vanes to move in the direction of the jet while under its influence, and also the quantity of water impinging to be equal to the nozzle discharge, by an analysis similar to that which has preceded, we obtain:

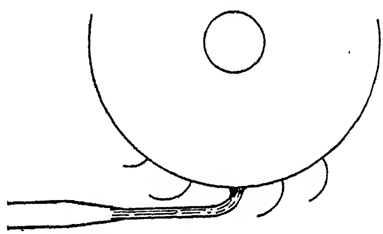


Fig. 23. Jet Striking Curved Vanes of a Revolving Wheel Tangentially.

Work and Efficiency. The work is:

$$\text{Work} = (1-\cos\theta)\, W\frac{(v-u)\,u}{g} \quad \ldots\ldots (41a)$$

This is zero when $u = 0$, or when $u = v$; also when $\theta = 0°$; and it is a maximum, and equal to:

$$\text{Work (Max.)} = \tfrac{1}{2}(1-\cos\theta)\, W\frac{v^2}{2g} \quad \ldots\ldots (42)$$

when $u = \frac{1}{2}v$.

The efficiency is: The efficiency is:

$$e = \tfrac{1}{2}(1-\cos\theta) \quad \ldots\ldots\ldots\ldots\ldots (43)$$

When $\theta = 0°$, the stream merely glides along the surface without doing work, and $e = 0$.

When $\theta = 90°$, the jet is deflected normally to the direction of motion, and,

$$\text{Work (Max.)} = \tfrac{1}{2} W\frac{v^2}{2g} \quad \ldots\ldots\ldots\ldots (44)$$

and efficiency is $e = \frac{1}{2}$, or 50 per cent, as for radial flat vanes.

When $\theta = 180°$, the stream is completely reversed, and

$$\text{Work (Max.)} = W\frac{v^2}{2g} \quad \ldots\ldots\ldots\ldots \quad \textbf{(45)}$$

in which case the efficiency is $e = 1$, or 100 per cent. The preceding analysis applies more directly to the case of a series of curved vanes moving in a straight line parallel to the jet, and coming in rapid succession under its influence. Such a motor is evidently impracticable.

In applying these considerations to water wheels, we must bear in mind that losses due to impact and friction have not been considered. The conclusions are therefore, to that extent, theoretic; but they represent limiting values which may be approached more and more closely, as the frictional and other resistances are reduced by means of correct design and construction. In the case of the conditions represented by Equation 45, since the efficiency is theoretically 100 per cent, it is clear that all the energy of the jet has been given up to the wheel, which would indicate that the absolute velocity of the water leaving the vanes must be zero; for if the water thus leaving has any absolute velocity, it still possesses some energy after passing clear of the wheel, which represents a portion of the original energy of the jet which has not been imparted to the wheel; the efficiency then could not be 100 per cent. This conclusion may be readily reached from the preceding analysis; for, since the best *absolute* velocity of the vane is $\frac{1}{2}v$, the water upon its surface has the *relative* velocity $v - \frac{1}{2}v = \frac{1}{2}v$, which is the same as the velocity of the vane, but in the *opposite direction;* then, if $\theta = 180°$, as in the case under discussion, the *absolute velocity of the water* as it leaves the vane, is $\frac{1}{2}v - \frac{1}{2}v = 0$.

While the above discussion shows that for maximum efficiency the velocity of the vanes should be one-half the velocity of the jet, the efficiency is not much lowered by slight variations of the vane velocity above or below the value indicated. It is also clear that to thus realize the full energy of the stream, we suppose the jet to both enter and leave the vanes in a direction tangential to the circumference, and a complete reversal is effected. It will be shown in a subsequent article that certain practical considerations render it impossible to fully realize these theoretic conditions.

If the vanes are plane radial surfaces, as in Fig. 21, the water passes from the wheel normally to the circumference, and

the highest obtainable efficiency is (theoretically) 50 per cent (Equation 35). In this case the water leaving the wheel still possesses absolute velocity to the extent of $\frac{v}{\sqrt{2}}$, the component of which, in the direction of motion of the vanes, is $\frac{1}{2}v$; this represents a dynamic pressure of $W\frac{\frac{1}{2}v}{g}$ pounds in that direction, or $W\frac{\frac{1}{2}v}{g} \times \frac{1}{2}v$ $(=P \times u)$ $= \frac{1}{2}W\frac{v^2}{2g}$ foot-pounds of work; that is, one-half of the original energy of the jet is carried away by the escaping water, and is thus lost to the wheel. Or, an absolute velocity of $\frac{v}{\sqrt{2}}$ represents kinetic energy to the amount of $\frac{W(\frac{v}{\sqrt{2}})^2}{2g} = \frac{1}{2}W\frac{v^2}{2g}$. Equation 58 shows even more clearly that in order to realize the full theoretic energy of the stream, the absolute velocity of the departing water ($v_1 = \frac{v}{\sqrt{2}}$ for this case) must be zero.

35. Case V. General Application. Usually the direction of motion of the vane is not the same as that of the jet. In Fig. 24, let the upper arrow v represent the direction of the jet as it impinges on the vane with an absolute velocity v; and let the arrow marked u represent the direction of motion of the vane, as well as its absolute velocity. While this case can be analyzed and solved in a manner similar to that employed in the preceding cases, it will be well here to adopt another procedure illustrating an important and useful principle:

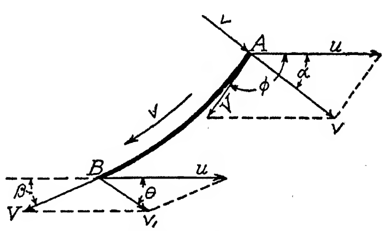


Fig. 24. General Case of a Jet Impinging on a Moving Vane.

The difference between the components of the absolute impulses of the entering and departing streams, in the direction of motion, is the resultant dynamic pressure in that direction.

Dynamic Pressure in a Given Direction. The absolute velocity of entry v being known, it remains to determine the absolute velocity of exit, v_1. By means of the principle enunciated in Article 29, we first find the relative velocity V with which the jet strikes the surface at A, by drawing to scale the lines v and u (both known) and

completing the parallelogram. V then represents, in intensity and direction, the relative velocity of the stream at A. The stream passes over the surface, and leaves it at B with this same relative velocity, if not retarded by friction or shock. Now, by the principle just referred to and used for the point A, the absolute velocity of the stream as it leaves the vane at B may be determined. Draw u and V, and complete the parallelogram; v_1 then represents the absolute velocity of the escaping water at B.

The absolute impulse of the stream before striking the vane at A is $W\frac{v}{g}$; its component in the direction of motion is $W\frac{v}{g}$ $\cos\alpha$. The absolute impulse of the stream as it leaves the vane at B is $W\frac{v_1}{g}$; its component in the direction of motion is $W\frac{v_1}{g}$ $\cos\theta$. Hence the dynamic pressure in the direction of motion is:

$$P = W\frac{v\cos\alpha - v_1\cos\theta}{g} \quad \ldots\ldots\ldots \quad (46)$$

This is a general formula for the dynamic pressure in any given direction exerted by a jet of water upon a vane moving in a direction parallel to a straight line, if α and θ be the angles between that direction and the directions of v and v_1.

If the surface is at rest, $v = v_1$, and Equation 46 becomes $P = (\cos\alpha - \cos\theta)\,W\frac{v}{g}$, which is Equation 27.

Usually, in the case represented by Fig. 24, the angles α and β are known, or assumed, and θ is unknown; it therefore becomes desirable to express the angle θ in other and known terms. By taking the components of the velocities at B in the direction of motion, it is evident that $v_1\cos\theta = u - V\cos\beta$; if this value be substituted in Equation 46, there will result:

$$P = W\frac{v\cos\alpha - u + V\cos\beta}{g}, \quad \ldots\ldots \quad (47)$$

in which,

$$V^2 = u^2 + v^2 - 2uv\cos\alpha \text{ (Trigonometry, from the triangle } A\,u\,v) \quad \ldots \quad (47a)$$

Curvature of Vane at Entrance. In order that the stream may strike the vane without shock, the curve of the vane at A should be tangent to the direction of V. It therefore becomes important to express the angle ϕ in known terms. From either triangle at A,

making use of the trigonometric principle that the sides of any plane triangle are proportional to the sines of their opposite angles, we obtain:

$$\frac{\operatorname{Sin}(\phi -)\,\alpha}{\operatorname{Sin}\,\phi} = \frac{u}{v} \quad \ldots\ldots\ldots\ldots (48)$$

which may be reduced, by known trigonometric relations, to:

$$\cot\,\phi = \cot\,\alpha - \frac{u}{v\,\sin\,\alpha} \quad \ldots\ldots\ldots (48a)$$

Equation 48*a* determines the angle ϕ, when u, v, and the angle a are known; and this fixes the proper curvature of the vane at the point A.

Example. In Fig. 24, let $u=70.71$, $v=100$, $\alpha=45°$, and $\beta=30°$. What is the dynamic pressure on the vane in the direction of motion, when 1 cubic foot of water strikes the vane per second? What should be the value of the angle ϕ in order that no loss by impact may occur?

From Equation 47*a*:

$$V = \sqrt{\overline{70.71}^2 + \overline{100}^2 - 2 \times 70.71 \times 100 \times .707} = 70.71 \text{ feet per second.}$$

From Equation 47:

$$P = 62.5\,\frac{100 \times .707 - 70.71 + 70.71 \times .866}{32.2} = 1{,}356 \text{ pounds.}$$

From Equation 48*a*:

$$\operatorname{Cot}\,\phi = 1 - \frac{70.71}{100 \times .707} = 0\,; \quad \therefore\ \phi = 90°$$

36. Case VI. Jet Impinging Upon Revolving Vanes of Water Motor. In the actual case of water motors, the vanes upon which the jet impinges revolve about an axis. The motion of every point on the vane is therefore circular; hence, at any instant, the direction of motion of any point is tangent to the circumference drawn through, or it is normal to the radius drawn to, that point. At any point, therefore, that portion of the dynamic pressure which is effective in producing motion is its component in the direction of motion of that point. Fig. 25 illustrates two cases of wheels with vertical axes, the vanes revolving in horizontal planes. In the one case (*b*), the water, after impinging, passes outward, or away from the axis; in the other (*a*), the stream passes inward, or toward the axis. The following analysis, however, is general, and therefore applies to both types. As heretofore, v and v_1 represent the absolute,

and V and V_1 the relative velocities of the entering and departing streams; u and u_1 (drawn normal to the radii r and r_1) represent the absolute velocities and directions of motion of the points A and B on the vane; the angles to be used in the analysis are sufficiently clear from the diagram, in view of what has preceded. Constructing the two parallelograms in the usual manner, there is obtained, at

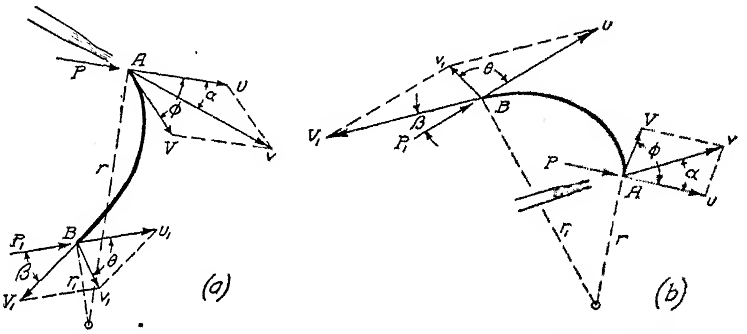


Fig. 25. Wheels with Vertical Axes, the Vanes Revolving in Horizontal Planes.

the point A, V as the relative velocity of the entering stream; and at the point B, v_1 as the absolute velocity of the departing stream. For the parallelogram at B, however, the value of V_1 must first be computed by means of Equation 54.

Components of Pressures in Direction of Motion. The total dynamic pressure exerted in the direction of motion will depend upon the impulses of the entering and the departing streams. The absolute impulse of the water on entering is $W\frac{v}{g}$; and that of the water on leaving is $W\frac{v_1}{g}$. The components of these in the directions of the motion of the vane at entrance and departure, are respectively:

$$P = W\frac{v \cos \alpha}{g} \text{ ; and } P_1 = W\frac{v_1 \cos \theta}{g} \quad \ldots\ldots (49)$$

Since their directions are not parallel, and the velocities of the points A and B are not equal, their difference cannot be taken to give the resultant dynamic pressure, as was done in Case V, which represented motion in a straight line; but this resultant pressure is not important. The two expressions in Equation 49, however, are useful in an analysis of the work that can be delivered by the vane.

37. Useful Formulæ. Since in any rotating body the linear velocities of points are directly proportional to their distances from the axis of rotation,

$$\frac{r}{r_1} = \frac{u}{u_1} \quad \ldots\ldots\ldots\ldots \quad \textbf{(50)}$$

The relative velocities V and V_1 are connected with the velocities of rotation by the following simple relation:

$$V_1^2 - V^2 = u_1^2 - u^2 \quad \ldots\ldots\ldots\ldots \quad \textbf{(51)}$$

Ordinarily, for a revolving vane, the data given or assumed will be the angles α, ϕ, and β; the radii r and r_1; the absolute velocity of the jet, v; the number of revolutions per second, n; and the weight of water delivered to the vane per second, W. Then,

$$u = 2\pi r n\,; \quad \text{and } u_1 = 2\pi r_1 n, \quad \ldots\ldots \quad \textbf{(52)}$$

from which u and u_1 may be determined.

In the triangle Auv (sides are proportional to sines of opposite angles),

$$V = \frac{v \sin \alpha}{\sin \phi}, \quad \ldots\ldots\ldots\ldots \quad \textbf{(53)}$$

which determines the relative velocity of entrance, V.

From Equation 51:

$$V_1 = \sqrt{u_1^2 - u^2 + V^2}, \quad \ldots\ldots\ldots \quad \textbf{(54)}$$

which gives the value of the relative velocity of exit, V_1. Finally, taking the components of the velocities at B in the direction of motion of that point, there results:

$$v_1 \cos \theta = u_1 - V_1 \cos \beta \quad \ldots\ldots\ldots \quad \textbf{(55)}$$

From the above equations, the numerical values of P and P_1 of Equation 49 can be fully determined.

Example. In Fig. 25; suppose $r = 2$ ft.; $r_1 = 3$ ft.; $\alpha = 45°$; $\phi = 90°$; $v = 100$ ft. per second; $n = 6$ revolutions per second. Compute the velocities u, u_1, V, and V_1.

From Equation 52:

$u = 2 \times 3.1416 \times 2 \times 6 =$ 75.4 feet per second.

$u_1 = \frac{3}{2} u$ $= 113.1$ " " "

From Equation 53:

$$V = \frac{100 \times .707}{1} = 70.71 \text{ feet per second.}$$

From Equation 54:

$$V_1 = \sqrt{113.1^2 - 75.4^2 + 70.71^2} = 110 \text{ feet per second.}$$

38. Work Derived from Revolving Vanes. In the discussion of "Work" and "Efficiency" under Cases IV and V, it was assumed that all points of the vane move with the same velocity; and in Case IV, that the stream enters upon it in the same direction as that of motion, or that $a = 0$. Considering the general case just discussed, it may be said that the work of a series of vanes arranged around a wheel may be regarded as that due to the absolute impulse of the entering stream in the direction of motion of the point of entrance, minus that due to the absolute impulse of the departing stream in the direction of motion of the point of exit; or,

$$\text{Work} = Pu - P_1 u_1 \text{ , } \quad \dots\dots\dots\dots (56)$$

in which P and P_1 are the components of the dynamic pressures due to the absolute impulses at A and B, in the directions of motion of the points A and B, respectively, as shown in Fig. 25 and Equation 49. Using the values of Equation 49, in Equation 56, there results:

$$\text{Work} = W \frac{u\, v \cos\ \alpha\ -\ u_1 v_1 \cos\theta}{g} \quad \dots\dots (57)$$

This is a perfectly general formula, applicable to the work of all wheels with outward or inward flow. It shows that the useful work consists of two parts—one due to the entering, and the other to the departing stream.

Another very simple general expression for the work of a series of revolving vanes may be deduced as follows: The total absolute energy of the entering stream is $W\frac{v^2}{2g}$; the total absolute energy of the departing stream is $W\frac{v_1^2}{2g}$; hence, neglecting friction and other resistances, the difference represents the energy imparted to, or taken up by, the wheel from the stream; that is:

$$\text{Work} = W\frac{v^2 - v_1^2}{2g} \quad \dots\dots\dots\dots (58)$$

which is a useful formula of wide applicability. From Equation 58, the efficiency is:

$$e = \frac{v^2 - v_1^2}{v^2} = 1 - \left(\frac{v_1}{v}\right)^2 \quad \dots\dots\dots (59)$$

Example. As a numerical example, consider the case of the outward-flow horizontal wheel driven by a jet from a fixed nozzle, shown in Fig. 26.

Let $r = 2$ feet;
$r_1 = 3$ feet;
$\alpha = 45°$ (approach angle);
$\phi = 90°$ (entrance angle);
$\beta = 15°$ (exit angle);
$v = 100$ feet per second;
$q = 2.2$ cubic feet per second;
$n = 337.5$ revolutions per minute.

It is required to find the useful work of the wheel, and its efficiency.

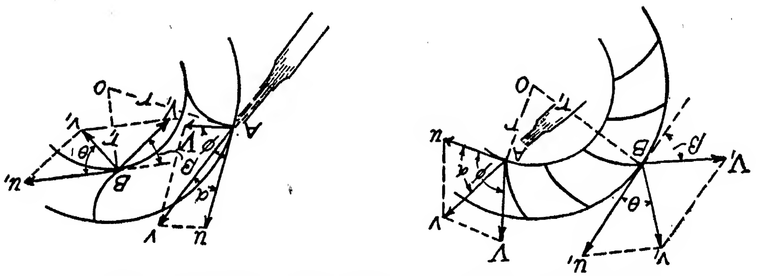


Fig. 26. Horizontal Wheels Driven by Jets from Fixed Nozzles.

From Equation 52:

$$u = 2\pi r n = 2 \times 3.1416 \times 2 \times \frac{337.5}{60} = 70.71 \text{ feet per second;}$$

and, from Equation 50:

$$u_1 = \frac{r_1}{r} u = \frac{3}{2} \times 70.71 = 106.06 \text{ feet per second.}$$

From Equation 53:

$$V = \frac{v \sin \alpha}{\sin \phi} = \frac{100 \times \sin 45°}{\sin 90°} = 100 \times 0.7071 = 70.71 \text{ feet per second.}$$

From Equation 54:

$$V_1 = \sqrt{u_1^2 - u^2 + V^2} = \sqrt{(106.06)^2 - (70.71)^2 + (70.71)^2} = 106.06 \text{ feet per second.}$$

From Equation 55:

$$v_1 \cos \theta = u_1 - V_1 \cos \beta = 106.06 - 106.06 \times \cos 15° = 3.61$$

Then, from Equation 57:

$$\text{Work} = 2.2 \times 62.5 \frac{70.71 \times 100 \times 0.707 - 106.06 \times 3.61}{32.2} = 19{,}712 \text{ ft.-lbs. per second.}$$

$$\frac{19{,}712}{550} = 35.8 \text{ horse-power.}$$

The theoretic energy of the jet is:

$$W\frac{v^2}{2g} = 2.2 \times 62.5\ \frac{(100)^2}{64.4} = 21,380 \text{ ft.-lbs. per second.}$$

$$\frac{21,380}{550} = 38.9 \text{ horse-power.}$$

Therefore the efficiency of the wheel is:

$$e = \frac{19,712}{21,380}, \text{ or } \frac{35.8}{38.9} = 92.2 \text{ per cent.}$$

This would seem to indicate a very high efficiency; but it must be borne in mind that losses in friction, shock, etc., have not been considered in the preceding analyses. The effect of such resistances will be to reduce the computed efficiency.

Example. In the above example, assume the same data, except that $\beta = 30°$.

The values of u, u_1, V, and V_1 are not altered.

$$v_1 \cos \theta = 106.06 - 91.85 = 14.21$$

and,

$$\begin{aligned} \text{Work} &= 14,910 \text{ ft.-lbs. per second,} \\ &= 27\ 2 \text{ horse-power.} \\ \text{Efficiency} &= 70 \text{ per cent.} \end{aligned}$$

In both of the above examples the work and efficiency may be simply computed from Equations 58 and 59, after the value of v_1 has been determined. From Fig. 24, parallelogram at B, since u_1 and V_1 are equal in the above examples, it follows that $\theta = \frac{1}{2}(180 - \beta)$; therefore, from Equation 55:

$$v_1 = \frac{u_1 - V_1 \cos \beta}{\cos \theta} = \frac{u_1 - V_1 \cos \beta}{\sin \frac{1}{2} \beta}$$

$$= \frac{106.06\ (1 - .966)}{.131} = 27.52 \text{ (for example 16);}$$

and,

$$v_1 = 106.06\ \frac{(1 - .866)}{.259} = 54.87 \text{ (for example 17).}$$

Substituting numerical values in Equations 58 and 59, the same results for the work and efficiency will be found as computed before.

FORMS OF APPLICATION

39. Hydraulic Motor. A *hydraulic motor* is a machine in which the energy stored in water is utilized to produce motion and thus perform work. The energy of water, as was explained in Article 5,

may exist in the form of gravity, of pressure, or of velocity; of these, gravity and pressure are not essentially or fundamentally separate and distinct phenomena, but rather the result of considering the weight of the water from different points of view. In general, then, it may be said that a hydraulic motor is an apparatus (usually a wheel) which is caused to move (usually rotate) by reason of a weight of water falling from a higher to a lower level, or because of the dynamic pressure induced by a change of direction, or of velocity, or both, in a moving stream. The dynamic pressure may be due to impulse, or reaction, or both. Many wheels are actuated by a combination in varying proportion of the above agencies, which are but manifestations of the energy existing in the water.

40. General Requirements for High Efficiency. The efficiency of any motor should, if possible, be independent of the quantity of water supplied to it; or, if the efficiency does vary with the supply, it should, when possible, be greatest in time of low water. It has already been shown that when W pounds of water fall through a height of h feet, or are delivered with a velocity of v feet per second, the theoretic energy in foot-pounds per second is:

$$K = Wh\,; \quad \text{or } K = W\frac{v^2}{2g}.$$

The actual work performed, or that may be performed, per second is equal to the theoretic energy, *minus* all the losses of energy. It is convenient to subdivide these losses into four general classes:

(*a*) Losses incidental to the conduction of the water from the supply to the motor, occasioned by friction and the various other resistances usually encountered, such as bends, changes of section, passages through orifices or other controlling devices which are not essentially parts of the apparatus itself, etc.;

(*b*) Losses in passage through the motor, which include friction, losses in eddies resulting from abrupt change in cross-section and improper entrance angle, and losses in passage through controlling devices which form part of the apparatus, etc.;

(*c*) The residual energy still possessed by the departing water flowing away with an absolute velocity v_1;

(*d*) Shaft and journal friction.

Sometimes the friction of the moving parts in the air or water is included, but will not here be considered.

41. Efficiencies. Let Wh' represent the energy lost in conduction; Wh'', that lost in passage through the wheel; $W\frac{{v_1}^2}{2g}$, the energy

still remaining in the departing water; and Wh''', the energy lost in shaft and journal friction; then,

$$k = W\left(h - h' - h'' - \frac{v_1^2}{2g} - h'''\right)$$

represents the actual useful work per second that the wheel is capable of performing. Accordingly, if v is the velocity due to the head h, the efficiency is:

$$e = \frac{k}{K} = 1 - \frac{h'}{h} - \frac{h''}{h} - \left(\frac{v_1}{v}\right)^2 - \frac{h'''}{h}.$$

This formula, being very general, leads to the four following broad statements of the conditions requisite for high efficiency:

(1) The water must be conducted to the motor, and

(2) The water must pass through the motor, with the minimum loss of energy.

(3) The water must reach the tail-race level with the minimum absolute velocity consistent with practical considerations, such as the necessity for quick and proper clearance of water from the buckets, etc.

(4) The friction and other mechanical resistances of the moving parts must be reduced to a minimum.

This analysis, with the corresponding formulae, compares the energy of the entire waterfall with the ultimate output of the machine. In estimating the power and efficiency yielded by the motor itself, *regarded as a user of water delivered to it with a definite amount of energy*, certain of the above losses should be omitted. Thus, losses in the conduction of the water to the motor cannot properly be charged against the motor; nor should losses in journal and shaft friction, which are outside and independent of the wheel regarded as a water user; in fact, the overcoming of journal and shaft friction is part of the work performed by the wheel, though it is not *useful* work. The energy in the departing water is properly chargeable to the wheel, since it is directly dependent upon the design or construction of the wheel. Therefore the hydraulic efficiency of the wheel may be stated thus:

$$e = 1 - \frac{h''}{h} - \left(\frac{v_1}{v}\right)^2; \quad \ldots\ldots\ldots\ldots \quad \mathbf{(60)}$$

or, as popularly stated, for high efficiency "the water should enter the wheel without shock, and leave without velocity." When the actual power and efficiency of a water motor are practically measured as described in Articles 88 *et seq.*, the shaft and journal friction

and air or water resistance are automatically included in the result. This explains why the results of actual tests of power and efficiency are always lower than the corresponding values completed from formulae derived without consideration of such losses. It is therefore well to employ two terms, *hydraulic efficiency* and *actual efficiency,* in order to distinguish clearly between the two sets of conditions involved.

42. Classification. In the absence of a uniform or generally accepted classification, hydraulic motors may be divided into two general classes:

(*a*) *Water-wheels,* in which the water does not enter and actuate the wheel around the entire circumference.

(*b*) *Turbines,* in which the water enters and actuates the wheel around the entire circumference.

Each of these main divisions has several subdivisions.

STANDARD WATER WHEELS

43. Overshot Wheel. In this form of wheel, the water enters at the top and acts mainly by its weight; nevertheless, in most forms, an appreciable amount of kinetic energy is likewise imparted to the wheel. Fig. 27 shows a vertical section of such a wheel. The *buckets* are formed by vanes or partitions made in two parts—one part *a* in line with the radius of the wheel, the other part *b* inclined in a direction definitely determined by the design. The bottom of the bucket is formed by the rim or *sole-plate F;* the side pieces are made by two *cheeks* or *shrouds E.* The whole is bolted to arms assembled on the hub, and supported by the axle.

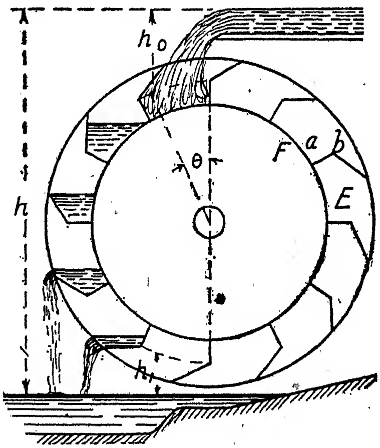


Fig. 27. Vertical Section of Overshot Wheel.

Analysis. Let *h* be the total fall from the surface of the water in the head-race or flume to the surface of the tail-race; and let *W* be the weight of water delivered to the wheel per second. The theoretic energy of the waterfall per second is *Wh* foot-pounds. The total fall *h* may be conveniently divided into three parts—namely,

h_0, the average head in filling the buckets; $h-h_0-h_1$, the average head of descent of the filled buckets; and h_1, that part of the head which remains between the empty buckets and the tail-race. The water strikes the buckets with a velocity v_0, approximately equal to $\sqrt{2gh_0}$; the buckets themselves are moving with a tangential velocity u approximately in the same direction as v_0; this occasions a loss of head in impact, h'' (Mechanics):

$$h'' = \frac{(v_0 - u)^2}{2g} .$$

The water then descends through the average distance $h-h_0-h_1$, acting by its weight alone; finally it drops out of the buckets, and reaches the level of the tail-race with the absolute velocity v_1, which represents part of the original energy wasted. Accordingly, the efficiency of the wheel is:

$$e = 1 - \frac{h''}{h} - \frac{v_1^2}{2gh} .$$

Since the water leaving the buckets has a velocity u when commencing the descent through height h_1, its velocity at the level of the tail-race is:

$$v_1 = \sqrt{u^2 + 2gh_1} .$$

Substituting the values h'' and v_1 in the equation of efficiency above,

$$e = 1 - \frac{v_0^2 - 2\,v_0\,u + 2\,u^2 + 2gh_1}{2gh} ,$$

and ascertaining by the usual procedure in such cases what value of u will render the efficiency e a maximum, it is readily found that:

$$u = \tfrac{1}{2}v_0 \; ; \quad \ldots\ldots\ldots\ldots \quad \textbf{(61)}$$

that is, theoretically, the velocity of the wheel should be one-half that of the entering water for maximum efficiency. With this value of u, the hydraulic efficiency is:

$$e\ (\text{Max.}) = 1 - \frac{1}{2}\frac{h_0}{h} - \frac{h_1}{h} \quad \ldots\ldots\ldots \quad \textbf{(62)}$$

and

$$\text{Work (Max.)} = Wh \times e = W\left(h - \frac{h_0}{2} - h_1\right) \quad \ldots \quad \textbf{(63)}$$

for the maximum efficiency and work of the overshot wheel. This equation teaches that one-half of the entrance drop h_0, and the whole of the exit drop h_1, are lost. Therefore, in order that the efficiency should be as high as possible, both h_0 and h_1 should be as

small as practicable. The former requirement may be met by making the wheel of large diameter; but h_0 can never be zero, for in that case no water would enter the wheel; practically the size of wheel is usually such that θ equals 10 to 15 degrees. The fall h_1 is made small by giving to the buckets such a form that the water will be retained as long as possible, and by having as little clearance as practically advisable between the lowest point of the wheel and the tail-race level. In the design illustrated in Fig. 28, the buckets are deep in order to hold the water as long as possible; and moreover, they are shaped to conform to the direction of the entering water, thereby avoiding shock. Wheels of this description have been constructed 50 feet in diameter. In this case the power is taken from the axle of the small pinion, which is driven by a toothed ring attached to the circumference of the wheel. In other cases the power may be taken directly from the shaft of the water-wheel, through intermediate gearing, or by a crank-shaft. The method of regulating the supply of water to the wheel is also shown in the diagram. The theoretic advantageous velocity of the overshot wheel was shown to be $u = \frac{1}{2}v_0$; practically, this advantageous velocity is found to be about $u = 0.4\,v_0$; and the efficiency of the wheel is high, ranging from 70 to 85 per cent, or over. One great advantage of the overshot wheel is that its efficiency is highest in times of drought, when the supply is low, for then the buckets are but partly filled, they do not begin to empty at as high a point above tail-water as when they are full; hence h_1 becomes small, with corresponding increase in efficiency. The main disadvantage of the overshot wheel lies in its size and its cost of construction. Moreover, its speed being slow (commonly from 3 to 6 feet peripheral velocity), it often requires the installation of somewhat complicated and expensive transmission gearing in order to drive machinery at a suitable speed; it is therefore best

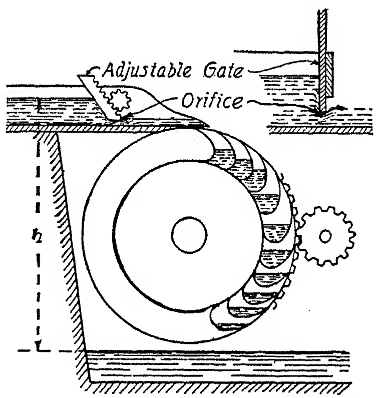


Fig. 28. Overshot Wheel with Deep Buckets to Hold Water as Long as Possible.

adapted to drive slow-moving machinery, usually with heads from 10 to 40 feet (though much larger heads have been used), and with a supply of from 100 to 350 gallons per second. A peripheral speed much greater than that commonly employed would result in a waste of water from the buckets due to centrifugal force.

The number of buckets and their depth are sometimes determined by formulæ, but they are largely matters of experience. If r is the radius of the wheel in feet, the number of buckets is usually $5r$ or $6r$, and their radial depth 10 to 15 inches. The width of the wheel parallel to the shaft is governed by the quantity of water actuating the wheel; it should preferably be so great that the buckets will not be quite full, thus reducing the fall h_1. If the tail-water level is constant, the lowest part of the wheel should be set just clear of that level; if it is variable, just sufficient clearance should be allowed to prevent interference and resistance in times of high water.

These precautions are necessary, for it is clear that the direction of motion of the buckets in the lowest portion of the wheel is opposite to the stream flow in the tail-race; and even slight submergence, therefore, will offer great additional resistance to its motion. This difficulty is sometimes obviated, when for any reason the wheel is to be submerged 4 or 5 inches (as by reason of variable tail-race level), by adopting a reverse-feed arrangement at the end of the supply channel, by which means the water is introduced on the back instead of on the front of the wheel, causing it to revolve in the opposite direction, so that the lower buckets move in the same direction as the tail-water. Such a wheel is often called a *back-pitch* or *back-shot* wheel.

For shallow streams of water with fairly constant depth, the supply channel is usually open-ended, as in Fig. 27; for deeper streams, or greater falls, the supply channel is provided with a sluice-gate or other regulating device, as in Fig. 28. Such a supply-regulating device is especially necessary in case of variable stream-flow.

Perhaps the largest overshot wheel in existence is that at Laxey, Isle of Man (Fig. 29), off the west coast of England. It is 72 feet 6 inches in diameter, and is said to yield 150 to 200 horse-power useful work, which consists in draining a mine 1,200 to 1,380 feet

deep. The supply of water for producing the power is conveyed to the wheel through an underground conduit, and from here it is carried up to the top of the masonry tower by the water pressure. At

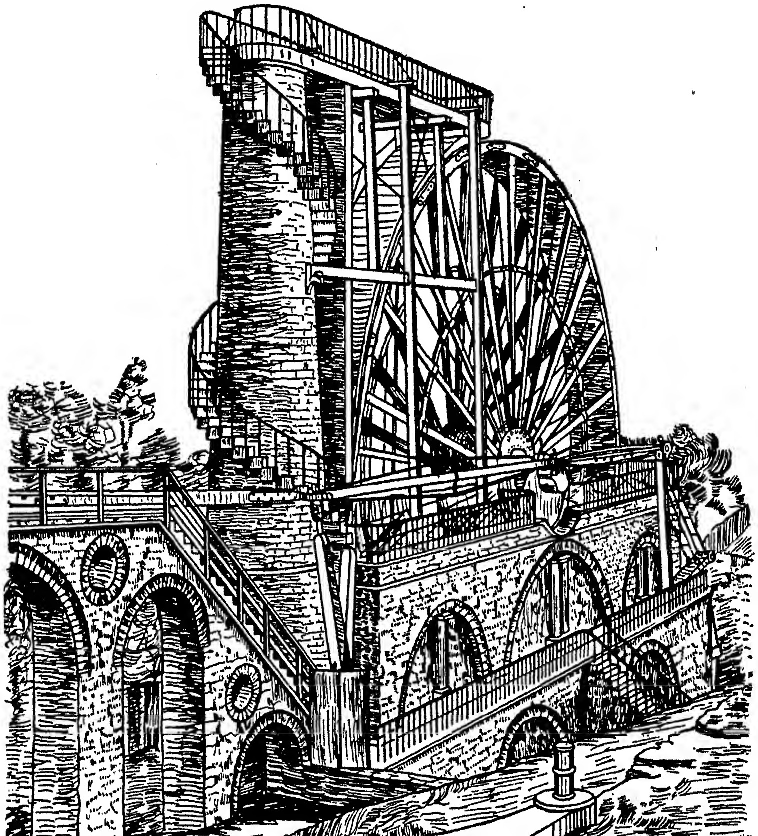


Fig. 29. Overshot Wheel at Laxey, Isle of Man.
Diameter of wheel, 72 ft. 6 in. Water carried up masonry tower by pressure, then flowing into buckets of the wheel.

this point it flows into the buckets of the wheel, the force of gravity furnishing the power to turn the wheel as the water passes to the bottom. Probably the largest wheel of this type in the United States was erected at Troy, New York, with a diameter of 62 feet and a width of 22 feet, developing 550 horse-power.

*TEST OF A STEEL OVERSHOT WATER WHEEL

A series of tests on a 10-foot steel overshot water wheel at the hydraulic laboratory of the University of Wisconsin has produced some interesting results.

A large number of American engineers consider the overshot water wheel as out of date—a relic of the past which has been replaced by the turbine, and for modern hydro-electric developments of large power there can be no question as to the superiority of the latter. The overshot wheel, however, is still capable of serving a distinct field, where the power to be developed is small and the speed of the machinery is slow and requires but little regulation. The fact that, after careful engineering investigation, wheels of this type are still being installed in Germany and France for small factories, pumping plants, and even for the development of electrical energy, may be taken as an indication of satisfactory and economical use of this type of motor.

44. Construction. The overshot wheel was formerly constructed of wood, and a number of picturesque and interesting examples of this type of construction still remain in this country. In recent years, however, the wheel has been built entirely of steel and iron. Although the initial cost of a wooden wheel is less than one built of steel, the latter possesses certain advantages which make it more desirable to install, notwithstanding the increased cost. Unless a wooden wheel be kept continually in operation, the constant swelling and drying of the wood soon causes all parts to get loose; the buckets leak and a considerable percentage of the available energy is wasted. In a steel wheel the buckets can be shaped more readily to accord with the design; a larger percentage of the total head is made available on account of the smaller thickness of the metal; and the friction in the bearings is decreased because a steel wheel is lighter than a wooden wheel well soaked with water. For these reasons the steel wheel is more efficient. Moreover, it is more durable, the cost of maintenance is less, and the parts are more readily assembled.

Fig. 30 shows an example of a steel overshot wheel built by the Fitz Water Wheel Company, Hanover, Pennsylvania—the wheel tested at the University of Wisconsin. Wheels of this type have been

*By C. R. Weidner, University of Wisconsin, *Engineering News*, January 2, 1913.

installed in various parts of the United States and South America. The buckets are formed by curved vanes attached to the cylindrical surface or inner circumference of the wheel, called the soling, and to the segmental circular housings on the sides. The vane thus forms the front or outer part of the bucket, the soling the back or inner side, and the housings form the sides. The buckets, soling, and rims are made of flanged steel riveted together. To the housings are bolted the radial arms, made of flat bar steel, which in turn

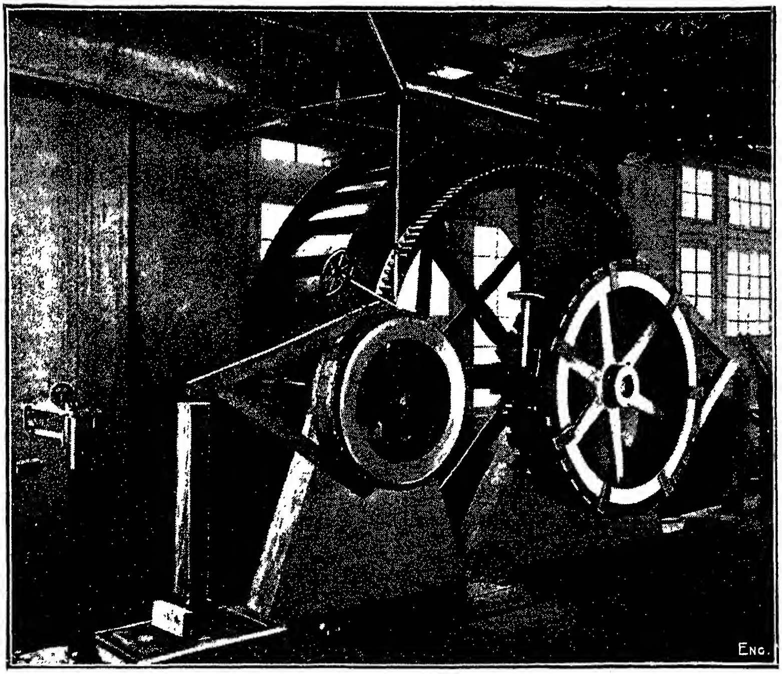


Fig. 30. Fitz Overshot Water-Wheel Undergoing Test
Courtesy of Fitz Water Wheel Company, Hanover, Pennsylvania

are bolted to cast-iron center flanges keyed to the shaft. Power is usually taken from the wheel by means of gearing and is transmitted by shaft, belt, or rope drive. In some designs the segmental spur gearing is bolted to the arms of the wheel, as in Fig. 31; in others a gear wheel is keyed directly to the shaft. Either ball or roller bearings are used.

45. Typical Advantages. The advantages of the overshot wheel are: (1) high efficiency; (2) adaptability to varying discharge;

(3) simplicity in construction; (4) reliability; and (5) no interference of operation on account of clogging with débris or ice.

Efficiency. Of these advantages the first two are of the greater importance. The results of the experiments discussed below show high efficiencies under a wide range of operating conditions. Reliable tests of turbines have been reported yielding as high as 89 per cent efficiency, but it is rarely that this figure is attained in an actual installation. In the smaller plants especially, where an overshot wheel would be capable of competing with a turbine, it is doubtful whether the turbines operate with an average efficiency

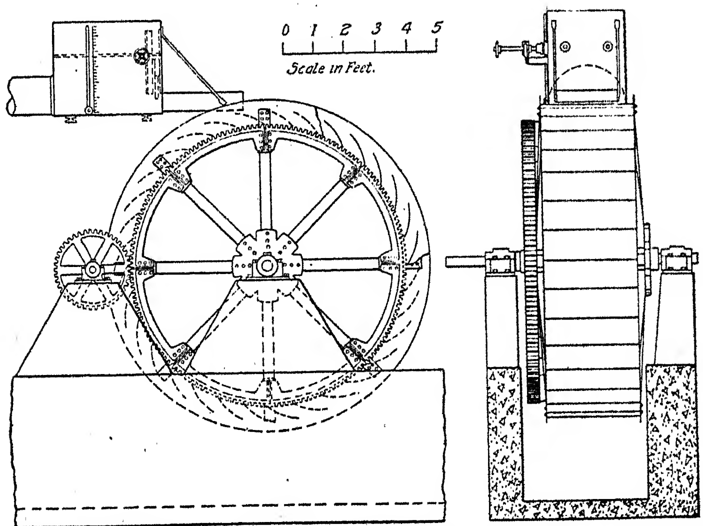


Fig. 31. End and Side Elevations of Fitz Overshot Water Wheel, Showing Construction

higher than 70 per cent. It is, however, extremely difficult to make any comparison as to the efficiency of two types of motors, unless both have been tested under exactly the same operating conditions. The installation of a turbine requires a technical analysis of the problem, and unless the turbine is set properly and selected for the particular conditions under which it is to operate, the efficiency will fall far below that of which it is capable when operating under the proper conditions. Very few of the small water powers developed receive the proper engineering supervision to promote a high degree of efficiency in their operation. The overshot wheel, on the other

hand, suffers but little from a lack of proper design or selection, although to obtain the highest efficiencies of which the wheel is capable, it is not advisable to dispense with a technical analysis.

Adaptability. To engineers familiar with the variation in efficiency of the turbine at part gate, a glance at the curves obtained from the Wisconsin experiments, Fig. 32, will be convincing as to the superiority of the overshot wheel, in respect to its adaptability to varying discharge. The experiments show that the range in the discharge may be as much as 400 per cent with only a difference of

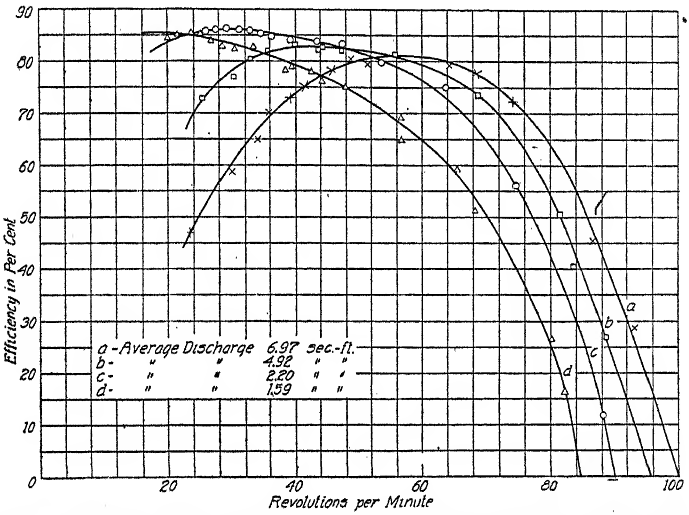


Fig. 32. Efficiency Curves of Fitz Water Wheel Obtained from Wisconsin Experiments

5 per cent in the efficiency of the wheel. The other advantages cited are of minor importance, though in particular cases they may influence the choice of type of motor to be selected.

46. Limitations of Type. The limitations or disadvantages are: (1) an economical limit in respect to the head and discharge to be developed; (2) a limit in respect to the speed of the machinery to be operated; (3) a limit in respect to the variation in the water levels in the head and tail races; and (4) larger space requirements than a turbine.

Head. On account of the increase in the weight of the wheel, as the head or discharge is increased beyond a certain limit, the cost

of the overshot wheel becomes so great that it can then no longer compete advantageously with the turbine. The economical field of the wheel, therefore, lies in developments which range approximately between 2 and 30 cubic feet per second discharge, with heads varying from 10 to 40 feet, corresponding to a maximum development of about 75 horse-power. Within this field, the question whether to install a water wheel or turbine must be decided on the basis of the particular conditions of the power to be developed, the class of machinery to be operated, and the cost of installation. Only a rough approximation can be made as to the cost of the two types of motors, from the figures furnished by the manufacturers. In general it may be said, that the initial cost of an overshot wheel, up to a diameter of 16 feet, will be about twice that of a turbine of equal horse-power, and above that diameter a little more than twice the amount.

Speed. The peripheral velocity of the overshot wheel, when operating efficiently, varies approximately between 3 and 7 feet per second, depending on the diameter, discharge, and velocity of the entering water. Hence, there would be a practical limit to the speed of the machinery to be served beyond which the loss of power through the necessary gearing would probably offset any gain in efficiency. It is, therefore, particularly adapted for the operation of slow-speed machinery, and should find a field of usefulness for the operation of small factories, for running grinders and other machinery on farms or country homes, and especially for pumping plants, where the pumps may be connected directly to the wheel shaft. For the operation of high-speed machinery, a loss of from 3 to 10 per cent may be estimated to occur through the necessary gearing or belting.

The principal application or use of water power at present is the development of electrical energy. In general, the opinion is held that only turbines on account of their high speed are applicable for this purpose. However, an increasing number of electrical plants, with water wheels as the motive power, have been built and are giving satisfactory service. The inertia of the heavy wheel and of the gearing provides a very uniform motion, and high efficiency at part load is a very desirable feature in electrical plants. It is a matter for the designing engineer to decide,

which type of motor will give the highest total efficiency for a plant of this kind.

Submergence. The efficiency of the overshot wheel drops off quite rapidly if the bottom of the wheel is submerged even but a few inches; therefore the wheel is poorly adapted to conditions where there is a considerable variation in the tail-water level. The turbine, however, will operate almost as efficiently when submerged as when not submerged and by use of the draft tube there is but little occasion to have it submerged. In this connection it should be noted that in periods of flood, when the water wheel is likely to be submerged, there is an excess of water, and as the wheel is particularly adapted for overload capacity the extra power developed would probably make up for any decrease in efficiency due to the submergence.

47. **Choice of Type.** Within these limitations the overshot wheel is capable of serving a field in water-power development and of competing successfully with the turbine both in efficiency and in cost of operation. The decision as to water wheel or turbine must be made only after careful investigation of the particular conditions to be met with—each type of motor should receive equal consideration. The considerations which influence this choice are: (1) head and discharge available, and probable variation in these; (2) cost of the motor, including setting and foundation; (3) relation of motor cost to available capital for plant construction; and (4) class of machinery to be served, and whether continuous or intermittent power supply is required.

Laboratory tests of a machine, when properly interpreted, undoubtedly have a great value; but it must be borne in mind that any test so made represents results under the exact conditions of the test. The conditions under which the Wisconsin experiments were performed approached practical conditions very closely. The wheel tested was of a standard pattern taken from the stock of the manufacturers. The structural features are simple and none of these features of the wheel itself were changed during the tests. Apart from the structural features of the wheel, the chief factors which influence the efficiency are the velocity of the entering water, the relation between this velocity and the peripheral speed of the wheel, and the discharge. The results should therefore be readily dupli-

cated in actual service if the wheel is set properly, and they may be taken as a guide in the design and installation of this type of water wheel.

48. Summary of Conclusions from Experiments. (1) The maximum efficiency obtained from these experiments was 89 per cent of the theoretical input delivered at the wheel shaft, or 85 per cent at the jack shaft.

(2) The transmission loss through the spur gear and pinion, including the friction in the bearings, reduced the efficiency of the wheel 3 per cent at maximum efficiency of operation. It was found that, within the limited range covered by these experiments, this loss increased with the speed and decreased with the horsepower transmitted.

(3) The efficiency of the wheel, other conditions remaining constant, increased with the decrease in the entrance velocity of the water. This statement is limited, however, by the fact that for maximum efficiency a definite relation exists between the peripheral velocity and the entrance velocity, so that a limit is reached when the entrance velocity equals or slightly exceeds the peripheral velocity.

(4) The variation in the discharge within reasonable limits, so that the coefficient of filling does not exceed approximately 0.50, has little if any effect on the efficiency of the wheel.

(5) For maximum efficiency of operation, the ratio of the peripheral velocity of the wheel to the velocity of the water entering the wheel is a constant, the value of which is approximately 0.90.

(6) Submergence of the wheel in the tail-water caused a serious decrease in the efficiency. A submergence of 3 inches caused a loss of 6 per cent at maximum efficiency of operation, which loss increased rapidly as the submergence was increased.

(7) For maximum efficiency, the point of impact of the water should lie as high in the wheel as possible. The point is regulated by the distance of the entrance spout or orifice from the wheel, and must be adjusted for each installation, otherwise a serious loss of energy occurs at entrance.

49. Breast Wheel. This type of wheel is designed to receive the water on one side, about or a little above the level of the hori-

zontal diameter; its lower portion, therefore, moves in the direction of the tail-water stream; for this reason the wheel may be *drowned*, or submerged, to a depth of 4 to 6 inches, which makes it suitable for use when both head-race and tail-race levels and supply are subject to variation. It is also evident from the manner of arranging the supply water, that this type is applicable only to small falls, from about 8 to 15 feet; for larger falls, the size of wheel would become impracticable. It is clear that the water acts both by impulse and by weight; therefore, to prevent the escape of the water before the buckets reach their lowest position, the lower quarter of the wheel is encased in a circular *breast* which encloses the buckets, thus practically compelling the water, or most of it, to remain therein until the lowest point is reached. In Fig. 33, water is conducted from the source in a channel or trough to and through an orifice A, which controls the supply to the wheel through regulation of the size of the orifice.

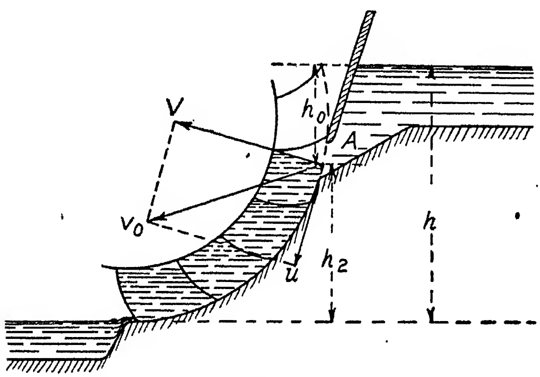


Fig. 33. Breast-Wheel with Supply Controlled through Size of Orifice.

In Fig. 34, the control of the supply is accomplished by means of a shuttle-gate arrangement which consists of a number of openings J J in the inclined end of the trough, one or more of which may be closed by shifting the sliding gate B. The guide-pieces are for the purpose of causing the water to enter the buckets in a direction most favorable for good efficiency. With the arrangement indicated in Fig. 34 considering the way in which the water enters the buckets, and observing that the mouths of the buckets are practically covered by the extension of the guide-pieces, it is evident that vents or air-holes F F in the sole-plate are necessary. Or the sole-plate may be dispensed with entirely, and the buckets formed of polygonal pockets, as b a c, in which the vents are naturally formed by the spaces left between the inner sides of consecutive buckets; these

being at the top, the buckets may be completely filled with water.

Work and Efficiency. In Fig. 33, the water is admitted through the orifice A, under a head h_0; it therefore strikes the wheel with a velocity v_0, which is approximately equal to $\sqrt{2gh_0}$, and actually equal to $c_1\sqrt{2gh_0}$, where c_1 is the coefficient of velocity for the orifice at A. The water, being then confined between the vanes and the curved breast, acts by its weight alone through the distance h_2, which is approximately equal to $h - h_0$; finally it escapes at the level of the tail-race with the velocity u, or the velocity of the circumference of the wheel. The reasoning in the article on overshot wheels may be applied to this case, by making the fall h_1 equal to zero, and the resulting conclusions may be considered to apply approximately to the case of breast wheels. Accordingly, the following relations are approximately true:

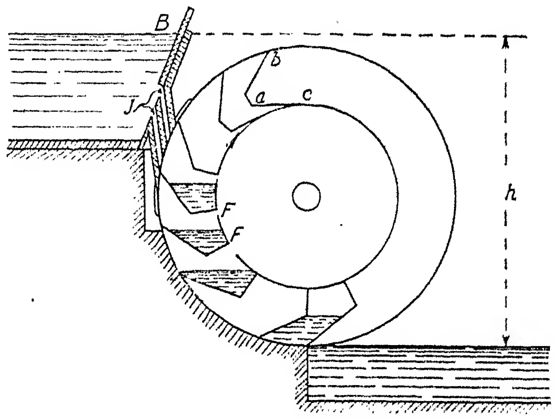


Fig. 34. Breast Wheel with Supply Controlled by Shuttle Gate.

The most advantageous theoretic velocity is

$$u = \tfrac{1}{2} v_0 = \tfrac{1}{2}\sqrt{2gh_0} \quad \ldots\ldots\ldots\ldots (64)$$

The maximum efficiency is theoretically:

$$e\,(\text{Max.}) = 1 - \tfrac{1}{2}\frac{h_0}{h} \quad \ldots\ldots\ldots\ldots (65)$$

The maximum work is theoretically:

$$\text{Work (Max.)} = W\,(h - \tfrac{1}{2}h_0) \quad \ldots\ldots\ldots (66)$$

Practically, the coefficient of velocity of the entrance orifice should be considered, as well as loss due to the clearance between wheel and breast, which will always exist; for any attempt to prevent this entirely by making the clearance less than about $\frac{3}{16}$ inch would

result in a considerable increase in circumferential friction, and also, if the wheel is slightly off center, in repeated shocks. For these reasons the efficiency of the breast wheel is materially less than that of the overshot wheel, the usual values ranging from about 50 per cent for small wheels to about 75 per cent for large, well-designed wheels.

When the fall is not great, the wheel is sometimes designed to receive the supply water at a point appreciably below the horizontal diameter; in this case it is frequently termed a *side wheel.* Its efficiency is lower than that of the regular breast wheel. The best wheels of this type have been constructed with diameters ranging between 12 and 24 feet, running with circumferential velocities between 6 and 10 feet per second. They may be regarded as a type intermediate between the regular breast wheel and the undershot wheel. Breast wheels are sometimes provided with some simple automatic governing device controlled by the speed of the wheel, whereby the feed-water orifice is partially throttled when the speed of rotation exceeds a definite predetermined amount.

50. Undershot Wheel. The common undershot wheel is provided with plane radial vanes, and the wheel is so set that the water impinges on the lower vanes only, in an almost horizontal direction. In one sense, then, the undershot wheel may be regarded as a special kind of breast wheel, which is operated entirely by the impulse of the moving water. The formulæ developed for the case of breast wheels may therefore be applied approximately to the case of undershot wheels by changing h_0 to h, and v_0 to v; thus, for the most advantageous velocity of the wheel:

$$u = \tfrac{1}{2}v = \tfrac{1}{2}\sqrt{2gh}; \quad \ldots\ldots\ldots\ldots (67)$$

the maximum efficiency is:

$$e = (\text{Max.}) = \tfrac{1}{2}, \text{ or 50 per cent}; \quad \ldots\ldots (68)$$

and the maximum work of the wheel is:

$$\text{Work (Max.)} = \tfrac{1}{2}Wh \quad \ldots\ldots\ldots\ldots (69)$$

Here, also, the coefficient of velocity of the water in passing through the orifice should properly be considered. In this type, as well as in the last, for reasons set forth in a preceding article, the maximum efficiency and maximum work are practically less than indicated in the foregoing formulæ; also, the most advantageous speed of the

wheel is more nearly $u = .40\sqrt{2gh}$ than $.50\sqrt{2gh}$. In practice the efficiencies of such wheels are found to lie between 20 and 40 per cent. The lowest efficiencies are obtained from wheels placed in an unconfined current of water, such as a wheel attached to a barge anchored in a stream; and the higher efficiencies may be expected from well-constructed wheels, in which the actuating stream of water is properly confined, so that it cannot spread laterally.

Fig. 35 shows a simple type of radial-vane undershot wheel operating under a head of water. Here it is seen that the wheel is set in a circular channel constructed with a radius a trifle larger than that of the periphery of the wheel. The sliding gate for regulating the supply from the penstock is arranged at an angle of about 45°, which enables its lower edge to be set close to the wheel rim. By this means the vanes are kept from contact with the moving water until they are almost vertical. The slight drop in the channel below the wheel compensates to some extent for the friction loss in passing the orifice of entry. The circular channel is succeeded by a gently inclined bed, so that the water maintains its uniform velocity after leaving the wheel, until, at a point well away from the wheel, the channel bed is given a sudden, steep inclination.

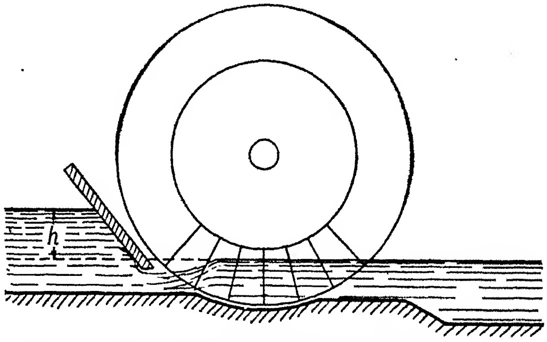


Fig. 35. Simple Type of Radial-Vane Undershot Wheel.

The depth of opening at the orifice usually varies from about 8 inches as a minimum, to about 20 inches in flood. The number of blades, N, is sometimes calculated from the empirical formula:

$$N = 4R,$$

in which R is the wheel radius. Then N and R will determine the spacing between the blades. In practice, this spacing may vary between 18 and 24 inches.

The undershot wheel is a relatively high-speed wheel; hence it

may be made more compact than the types described before; its construction and installation are extremely simple, and, from these points of view, it is economical. But its efficiency is lower than that of the other types; it is suitable only for very simple installations, to drive machinery at relatively high speed, where an ample supply of water is available, under a low head.

51. Poncelet Wheel. In this wheel (Fig. 36), the vanes are curved in such a way that the water enters through the regulating orifice or opening without shock. Let v be the absolute velocity of the entering stream, and u the peripheral speed of the wheel. The stream, entering with the absolute velocity v, impinges tangentially on the smooth vanes, which are themselves moving in the same general direction with an absolute velocity u. The relative velocity of the water is therefore $v-u$; and it glides smoothly up the curved vane in the general direction of motion of the stream to a height corresponding to this velocity; when at its uppermost point, it is at rest relatively to the vane; it then falls, exerting pressure as it falls, gliding along the vane in the general direction opposite to the motion of the stream, attaining the velocity $v - u$ at the lowest point or extremity of the vane, and passing from the vane tangentially. Its dynamic pressure is therefore due to both impulse and reaction:

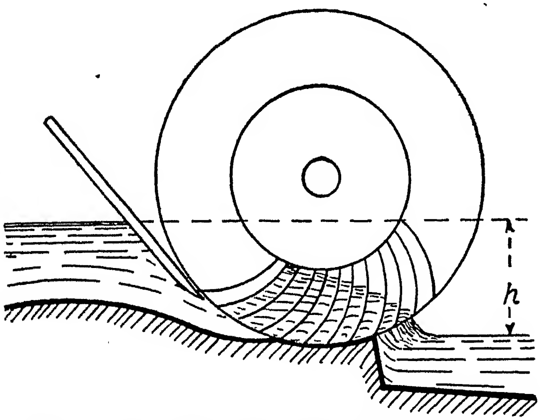


Fig. 36. Poncelet Wheel, Water Entering without Shock.

$$P = F + R = 2F = 2W\frac{v-u}{g};$$

the work of the wheel is $(P \times u)$;

$$\text{Work} = 2W\left(\frac{v-u}{g}\right)u;$$

and this is a maximum, and equal to:

$$\text{Work (Max.)} = W\frac{v^2}{2g} = Wh \quad \cdots\cdots\cdots \quad \textbf{(70)}$$

when $u = \frac{1}{2}v$.

Since the theoretic energy of the stream is $W\frac{v^2}{2g} = Wh$,

$$e\ (\text{Max.}) = 1, \text{ or } 100 \text{ per cent} \quad \ldots\ldots\ldots (71)$$

This follows from the fact that with the advantageous velocity $u = \frac{1}{2}v$, the absolute velocity of exit is zero; hence the stream "enters without impact, and departs without velocity."

The preceding analysis and the conclusions are theoretic, since they do not consider the various losses of head or energy which must take place. Practically, the efficiency lies between 65 and 75 per cent.

The curved form is given to the bed of the channel of approach, in order to direct the entering stream of water so as to avoid shock. The depth of the vane should be such that the entering water may run up its length (due to its relative velocity) without interference. The spacing of the blades usually ranges between 10 and 18 inches.

The Poncelet wheel, like other undershot wheels, has a relatively high speed; its efficiency is almost independent of the flow, and also of the speed, when a curved channel of approach is used. Moreover, this speed does not vary much, in spite of considerable variations of head. This form of wheel may be used to advantage with a head not exceeding about 6 feet, when the application of power does not call for a high velocity, as for pumping, grinding, etc.

52. Independent Deductions. In the foregoing cases, the deductions, largely by comparison and analogy, resulted in conclusions more or less approximately true. In each case, however, these relations may be developed quite independently, giving theoretically accurate results. For example, take the case of the breast wheel represented in Fig. 33. In the figure, let Av_0 and Au represent in intensity and direction the velocities, respectively, of the entering water and of the vanes, inclined to each other at an angle α. The dynamic pressure exerted by the water on the vanes, in the direction of motion, is:

$$P = W\frac{v_0 \cos \alpha - u}{g};$$

and the work per second is:

$$K = W\frac{(v_0 \cos \alpha - u)}{g}u. \quad \ldots\ldots\ldots (72)$$

The work K, of the dynamic pressure alone, is a maximum, and equal to:

$$\text{Work (Max.)} = W\frac{v_0^2 \cos^2 \alpha}{4g} \quad \ldots\ldots\ldots (73)$$

when $u = \frac{1}{2}v_0 \cos \alpha$.

To this value of K must be added the term Wh_2, representing the work done by the weight of water in the buckets falling the distance h_2; this term is theoretically independent of the speed; accordingly,

$$\text{Total work (Max.)} = W(\frac{v_0^2 \cos^2 \alpha}{4g} + h_2) \; ; \quad \ldots\ldots (74)$$

but $v_0 = c_1\sqrt{2gh_0}$, where c_1 is the coefficient of velocity for the orifice at A. Therefore,

$$\text{Total work (Max.)} = W(\frac{1}{2}c_1^2 \cos^2 \alpha .\ h_0 + h_2) \; ; \cdot \quad (75)$$

and the maximum hydraulic efficiency is:

$$e\,(\text{Max.}) = \frac{1}{2}c_1^2 \cos^2 \alpha\ \frac{h_0}{h} + \frac{h_2}{h} \cdot \ldots\ldots\ldots (76)$$

If, in these equations (73, 75, and 76), h_2 be replaced by its equal $h - h_0$, and if c_1 equals unity, and the angle α equals zero, there will result the approximate equations 64, 65, and 66, deduced in Article 49.

The angle α, however, cannot be zero; in fact it cannot practically be made less than about 10 degrees, for then little or no water would enter the wheel; it should, nevertheless, be as small as practicable, and is usually found between 10 and 25 degrees. The value of the coefficient c_1 is rendered large by well rounding the edges of the orifice; in this way c_1 may be made equal to .95 or even .98. In a manner similar to the above, formulæ for the other cases discussed may also be developed, with a greater degree of accuracy, theoretically considered. It is evident, however, that the approximate formulæ are sufficiently exact for most purposes, since the losses due to improper entry, foam, and leakage, cannot be algebraically expressed.

SPECIAL FORMS OF WHEELS

Water wheels in great variety have been in use from very early times, some of them operating with a fair degree of efficiency. A few of these forms will be very briefly described.

53. Sagebien Side Wheel. The buckets of this wheel (Fig. 37) are formed by flat vanes which are tangent to the horizontal cylinder O, whose axis is concentric with the shaft of the wheel. The depth of the bucket-ring is relatively large, and there is no sole-plate, each bucket forming a sort of vessel open on top and bottom. The wheel turns in a circular channel, prolonged upstream by a suitable iron casing, sometimes called a *swan's neck*. The side cheeks of the

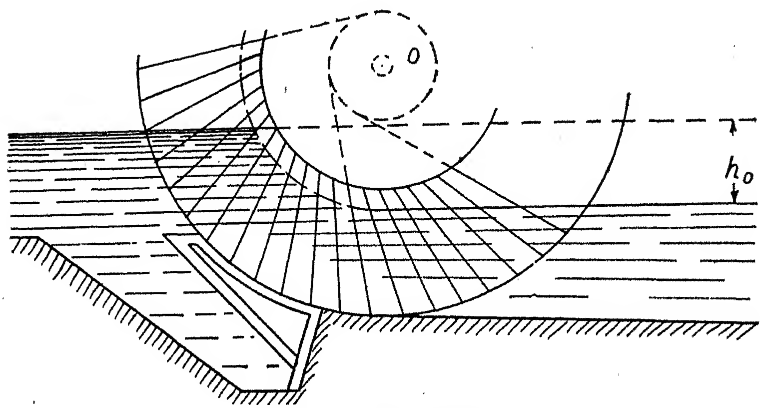


Fig. 37. Sagebien Side Wheel.

channel are continued downstream to the point where the wheel leaves the tail-race. There is very little work done by the water on the vanes beyond a point vertically below the center of the wheel. The inclination of the blades is not favorable for their easy emergence on the downstream side; but, as the speed of the wheel is rarely so great as 3 feet per second, being more usually between 1 and 2 feet, this resistance is small. The efficiency of this wheel, on account of its low speed (since resistances increase more or less rapidly with the speed), is very high, ranging from 80 to 90 per cent according to the height of the fall and the diameter, which varies between 20 and 40 feet, depending upon the fall available, the variability of the supply, and the fluctuation in the tail-race level. The number of revolutions per minute is often less than 1, and rarely exceeds 2½. The penstock speed is usually 1 to 2 feet per second, and this is about the velocity with which the water enters the wheel. The spacing of the blades, measured on the outside of the wheel is about 15 inches. This type of wheel is used for small falls, from 2 to 9 feet, and is suitable for large

flows. On account of its slow speed, it is adaptable only for installations where the machinery runs slowly and opposes uniform resistance to driving.

54. Millot Wheel. This is a form of breast wheel (Fig. 35) in which the breast is not needed. The supply channel divides into two branches, which pass around to the inner side of the wheel, so that the water enters at the inner circumference. This wheel is difficult to construct, and can be used only for small powers, since, by reason of the feed-water arrangement, the arms must be placed in the middle section of the wheel, instead of being fixed to the flanges; for this reason the breadth is limited to about 5 feet.

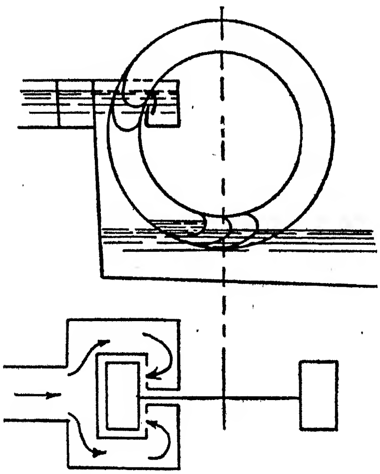


Fig. 38. Millot Wheel.

55. Floating Wheel, or Current Wheel. This type is simply an undershot wheel with flat, radial vanes, erected on a scow or barge intended to be anchored in a stream, or mounted on some suitable framework built up from the stream bed. The flat blades are attached to an inner circle, but are not enclosed in shrouds, so that the water has very free entry. As the barge rises and falls with the changes of the stream level, the depth of blade immersion is constant. The efficiency is theoretically a maximum, and equal to 50 per cent, when the peripheral speed of the wheel is one-half the velocity of the current; actually, it rarely reaches 40 per cent. When such a wheel is required to drive stationary machinery—that is, machinery so mounted that it does not follow the fluctuations in the surface level—some special device must be employed to insure the required condition of constant depth of paddle immersion. These wheels are extremely simple, but require to be of large size in order to develop even a moderate amount of power.

Wheels of this type have been used for operating dredges on the river Rhine, Germany; they have also been used to a limited extent, principally for irrigation purposes, in the western part of the

United States. One at Fayette Valley, Idaho, was said to be 28 feet in diameter, with 28 paddles, each 16 feet long and $2\frac{1}{2}$ feet wide.

56. Tympanium. This is an ancient form of circular open-frame wheel (Fig. 39), fitted with radial partitions so directed as to point upward on the rising side of the wheel, and downward on the descending side. The wheel is mounted in such a way that its lowest parts are submerged to a convenient depth, and it may be turned by the impulse of the current impinging on radial vanes arranged around its circumference. The partitions scoop up a quantity of water, which, as the wheel revolves, runs back toward the axis, where it is discharged into a trough that conveys it away. A very evident disadvantage of this form of wheel is the fact that the water has to be

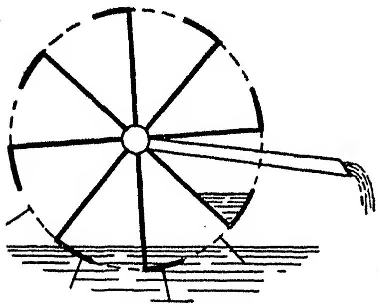


Fig. 39. Tympanium.

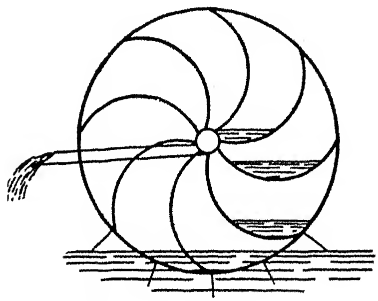


Fig. 40. Scoop Wheel.

raised at the extremity of each radius, so that its lever arm, and therefore its resistance, increases as it is raised to a horizontal plane. This defect does not exist in the next type.

57. Scoop Wheel. As this wheel (Fig. 40) revolves, the partitions dip into, and scoop up the water; and as they ascend, the water is discharged into a trough placed under one end of the shaft, which is arranged in as many compartments as there are partitions or scoops.

An improved form of scoop wheel is shown in Fig. 41, which consists of four curved scrolls or channels suitably mounted on the wheel body. The water is conveyed to the central chambers by the scrolls, and it then flows away in a channel or trough.

Many other forms of water motor might be shown, most of them ancient and obsolete, which were mainly used for the purpose of raising water; but the above examples serve to indicate some of the principal devices employed for the purpose.

58. Wave Motors. Many attempts have been made to develop useful power from the almost ceaseless motion of the ocean waves. The essential mechanism usually consists of some form of float which is constrained by a fixed shaft, or a series of such shafts, fastened to a suitable foundation, to move in a vertical direction under the influence of the motion of the waves. The float, by its motion, operates a system of levers and wheels, or ropes and pulleys, which may be made by suitable connections to compress air, or to raise water from a lower to a higher level. In some such way, the irregular or intermittent character of the wave motion may be made to store up power, which, in turn may be released uniformly. Fig. 42 is a diagrammatic representation of such a device.

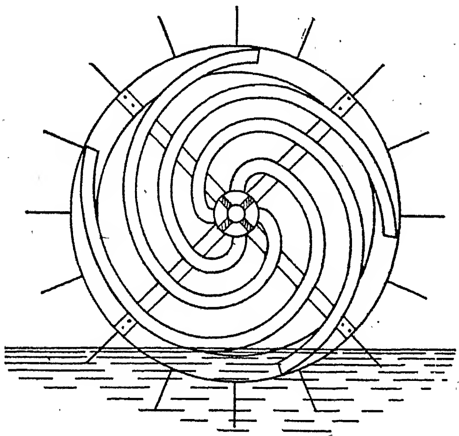


Fig. 41. Scoop Wheel, Improved Type.

59. Tide Wheels. Ocean tides furnish a reliable means of developing power under suitable conditions. Particularly in the vicinity of tidal rivers, and more rarely along shore, the physical configuration of the land may afford opportunity for impounding large volumes of water during the rising of the tide, which may be made to develop power at ebb by flowing out through a suitable channel and operating one or more wheels. Since the wheels must necessarily remain idle during the rising of the tide, some suitable means must be provided for storing power, so that the machinery dependent upon this power may be in continuous operation, or may operate at any time, irrespective of the tidal conditions. Where power is used intermittently—as in some pumping plants which operate only a certain number of hours each day of 24 hours— a system of power storage, while convenient and advisable to provide against the contingency of a breakdown or other mishap, is not so necessary.

60. Water-Pressure Engine. The water-pressure engine

performs work by reason of the static pressure of water acting upon a piston or a revolving disc. The cylinder and piston type of motor has a reciprocating motion identical with that of the steam engine; and the operation is very similar, the water entering and leaving through ports which are opened and closed by valves properly connected with the piston-rod. The useful work is due to the difference in the pressure of admission and discharge. As in the case of the steam engine, the reciprocating motion is generally changed by suitable mechanism into rotary motion before being applied to drive machinery. In the other type, the rotary motion is obtained directly from the shaft of the rotating discs or vanes. This latter type has not been widely used, as in practice there are many inherent difficulties in this mode of transmitting high power.

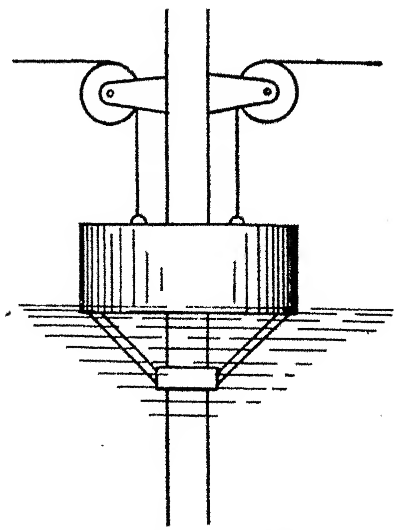


Fig. 42. Device for Utilizing Power of Wave Motion.

IMPULSE WHEELS

The term *impulse wheel* is sometimes used to include only those special forms of hydraulic motor which are driven by a jet of water issuing from a nozzle and impinging upon vanes or buckets of special shape attached to the circumference of the wheel. This definition would improperly exclude such motors as the undershot wheel, which is nevertheless a true impulse wheel actuated by a broad stream of water; as well as other types of true impulse wheels.

HORIZONTAL WHEEL TYPES

61. Conditions for Efficiency. When a wheel operated by a stream of water issuing from a nozzle and impinging on its vanes is so placed that its plane of rotation is horizontal (the axis being vertical), it is called a *horizontal impulse wheel.*

There are two general classes of such wheels, the *outward-flow*, and the *inward-flow*, as described in Article 36 and illustrated dia-

grammatically in Figs. 25 and 26. In order to deduce the conditions or relations for maximum efficiency, consider Fig. 26; in which both types are represented, so that the following analysis and the resulting conclusions will be generally applicable to such wheels. The construction of the parallelograms, and the notation, being the same as heretofore, further explanation will be unnecessary.

In order that the water may enter the wheel without shock or foam, the relative velocity V should be tangent to the vane at A as explained before. This condition of tangency will obtain when u and v are proportional to the sines of their opposite angles, in the triangle Auv (as in Article 51, Equations 48 and 48a); that is:

$$\frac{u}{v} = \frac{\sin(\phi - \alpha)}{\sin \phi}; \text{ or, } \cot \phi = \cot \alpha - \frac{u}{v \sin \alpha}$$

The absolute velocity of exit v_1 should be very small (Equations 58 and 59), for the energy represented by this velocity is not given to the wheel, but wasted. Theoretically it should be zero for maximum efficiency, as has already been shown; but practically, if this were the case, the vanes would be unable to clear themselves of the contained water. This absolute velocity v_1 will be small when

$$u_1 = V_1 \quad \ldots\ldots\ldots\ldots \quad (77)$$

These two equations are usually given as representing the conditions of maximum hydraulic efficiency. Equation 77, however, is only approximately true, the real minimum value of v_1 is found when $V_1 = u_1 \cos \beta$, in which case $v_1 = u_1 \sin \beta$; but this equation leads to very complex formulæ. Hence the simpler relation of Equation 77, which is sufficiently accurate, will be used.

Referring to Equation 51, it is clear that if u_1 equals V_1, u must equal V. Then, from the parallelogram at A, Fig. 26, it is seen that when $u = V$, the diagonal bisects the angle ϕ; or,

$$\phi = 2a \quad \ldots\ldots\ldots\ldots \quad (78)$$

Using this value of ϕ in Equation 48, there results:

$$u = \frac{v}{2 \cos \alpha} \quad \ldots\ldots\ldots\ldots \quad (79)$$

Equations 78 and 79 state the conditions involved in Equations 48 and 77, for maximum hydraulic efficiency, in terms sometimes more convenient for use. When a wheel constructed according to this

condition (Equation 78) is running with the advantageous velocity u of Equation 79, the absolute velocity of exit is:

$$v_1 = v \frac{r_1}{r} \frac{\sin \frac{1}{2} \beta}{\cos \alpha} ; \quad \ldots\ldots\ldots\ldots (80)$$

and the corresponding hydraulic efficiency (Equation 59) is:

$$e = 1 - \left(\frac{r_1}{r} \frac{\sin \frac{1}{2} \beta}{\cos \alpha} \right)^2 \quad \ldots\ldots\ldots (81)$$

An analysis of this formula teaches that, for high efficiency, both the approach angle α and the exit angle β should be small; but they cannot be zero, otherwise water would not pass into and out of the wheel. Values of 15 to 30 degrees are common. Since, for small angles, the sine varies much more rapidly than the cosine, the equation of efficiency also shows that β is more important than α; so that if β be very small, α may be as large as 40 or 45 degrees, with high efficiency. The equation further shows that for given values of α and β, the inward-flow wheel, in which r_1 is less than r, has a higher efficiency than the outward-flow wheel.

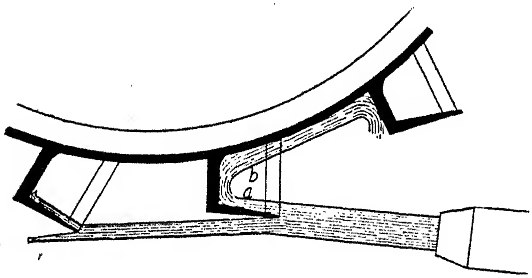


Fig. 43. Faulty Design of Vane.

The actual curve between the entrance and exit points of a vane is not of importance, provided it be smooth and gradual, as abrupt changes of direction lead to shock and loss of energy.

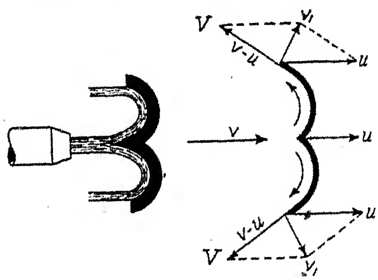


Fig. 44. Good Type of Vane, with Double Cups and Dividing Central Rib.

VERTICAL WHEEL TYPES

62. Commercial Tangential Forms. Of this type of wheel (frequently called a *hurdy-gurdy* when the vanes are flat planes) there are several forms in the market which have been highly developed and

which differ principally in details; these are known by various trade names, such as *Pelton-Doble, Cascade,* etc. Essentially this type

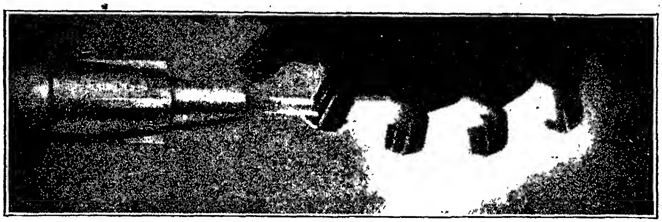


Fig. 45. Instantaneous Photograph of a Water Jet Striking the Buckets of a Pelton Motor

consists of a wheel mounted on a horizontal shaft, which transmits the power received from a jet or several jets of water acting upon a series of cup-shaped vanes attached to its periphery. The simplest type would be a wheel with flat radial vanes, as in Fig. 21; but, as has already been shown, the efficiency in such a case would be low, so that in practice curved vanes are invariably used.

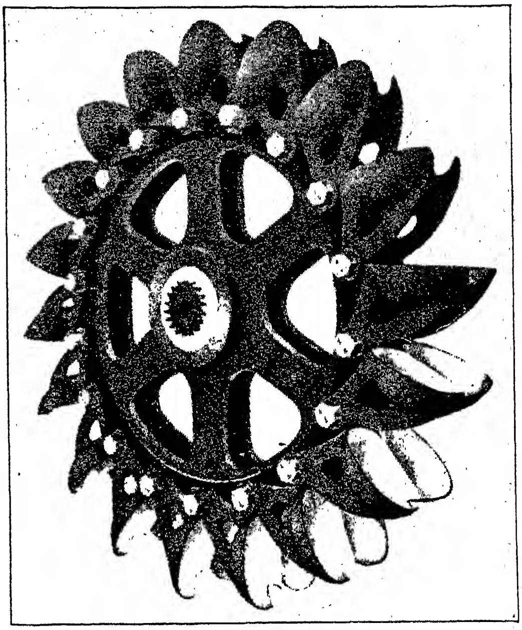


Fig. 46. Single Tangential Water Wheel with Doble Ellipsoidal Buckets
Courtesy of Pelton Water Wheel Company.

In Fig. 43 is shown a faulty design of vane, for the water, after striking the outer lip, is abruptly changed in direction at the corners *a* and *b*, with consequent shock and loss of energy; also, after leaving a vane, the stream strikes the back of the one adjoining, thus producing back-pressure, with further loss of energy. For these reasons the cups or vanes must be very carefully designed.

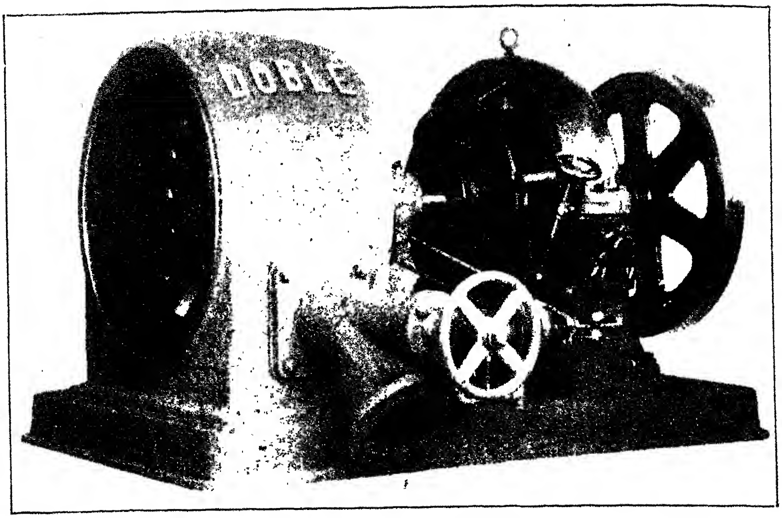


Fig. 47. Automatically Governed Pelton-Doble Tangential Water Wheel Driving an Exciter Generator

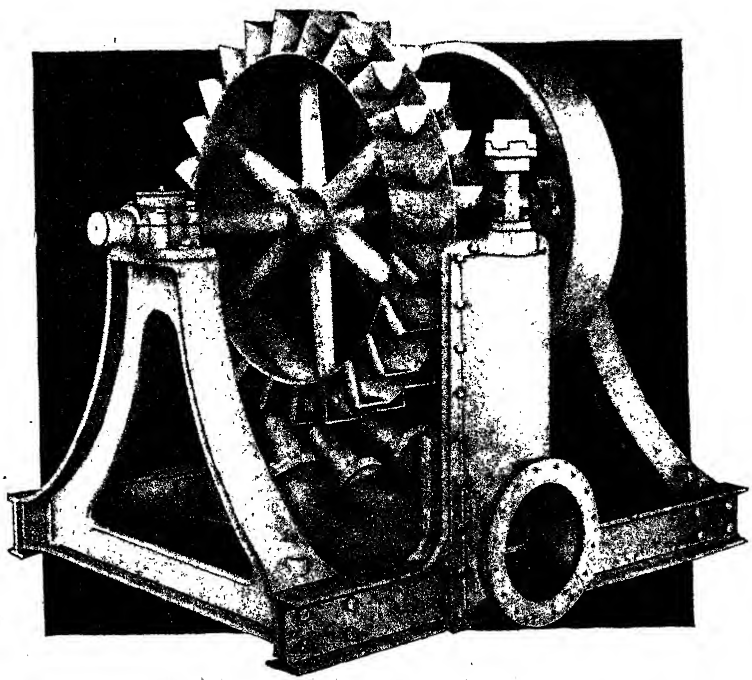


Fig. 48. The "Cascade" (Leffel) Impulse Wheel

In the best forms, the vanes are double cups or buckets with a central rib designed to divide and turn the stream sidewise, while at the same time deflecting it backward, opposite the direction of motion, Fig. 44. The effect of the stream is shown in Fig. 45.

Fig. 46 shows the usual method of attaching the buckets to the wheel; it is clear that in these designs one or more buckets may be very easily and quickly removed and replaced when this is rendered necessary by reason of wear or breakage.

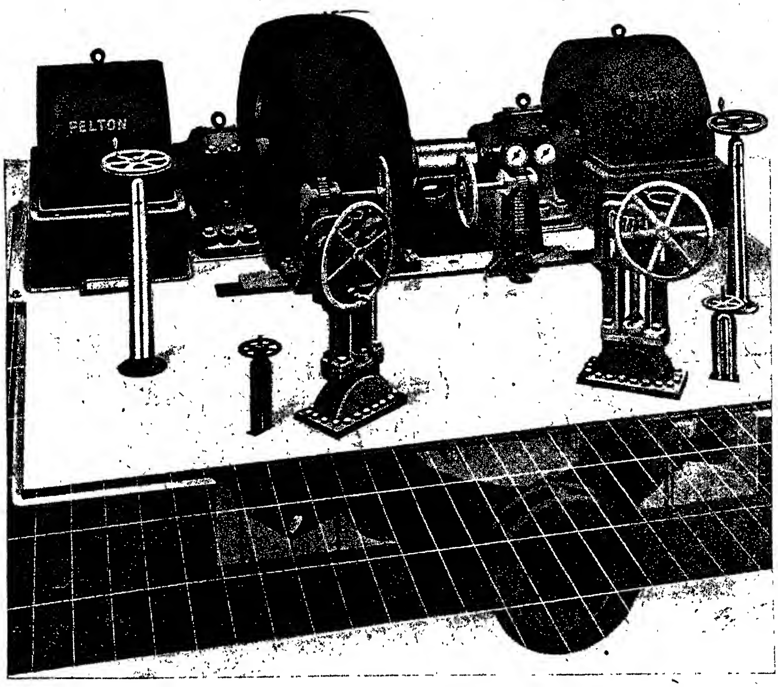


Fig. 49. Typical Double "Overhung" Unit Driving Generator
Courtesy of Pelton Water Wheel Company, San Francisco, California.

In Fig. 47 is shown an automatically governed Pelton-Doble tangential water-wheel driving an exciter generator. The speed of the machine is automatically regulated by a governor designed to operate by either oil or water pressure.

In the *Cascade* (*Leffel*) wheel (Fig. 48), the *lobes* or half buckets are set *staggering*, or *breaking joint*, on opposite sides of a thin circular disc, the sharp edge of which serves the same purpose as the central rib of the other forms in dividing the stream.

63. Characteristics. The analysis and conclusions of Article 34, Fig. 23, apply in the case of these wheels; namely, the most advantageous velocity, theoretically, is:

$$u = \tfrac{1}{2} v;$$

and at this velocity, the efficiency is a maximum, and equal to;

$$e \text{ (Max.)} = 1, \text{ or } 100 \text{ per cent,}$$

when $\theta = 180°$—that is, when the stream is completely reversed. However, θ cannot be made equal to 180°, so as to completely reverse the direction of the stream, without interference between the departing water and the adjoining vane, as shown in Fig. 43, where the water is deflected vertically; and this is equally true when the

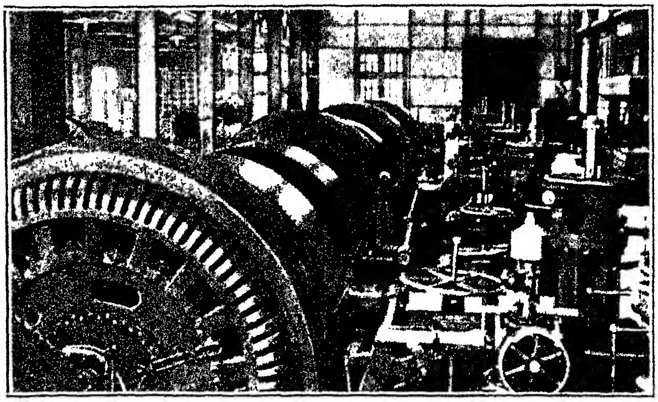


Fig. 50. Plant of Sierra and San Francisco Power Company, Showing Four 16,000 H. P. Tangential Wheel Units
Courtesy of Pelton Water Wheel Company, San Francisco, California.

stream is deflected sideways. The vane is therefore so shaped as to throw the divided stream just clear of the next vane, which condition makes it necessary that θ shall be less than 180 degrees, and consequently the efficiency will be less than 100 per cent, even theoretically. Nevertheless this form of wheel probably comes as near as any to realizing the theoretic condition for maximum efficiency.

As in all the other cases discussed, the theoretic conclusions derived from analyses are not quite true practically. Thus the most advantageous velocity of the wheel is somewhat less than .5 of the jet velocity (though it is probably always considerably greater than .4 that velocity), while the maximum efficiency may be 90 per cent or somewhat higher.

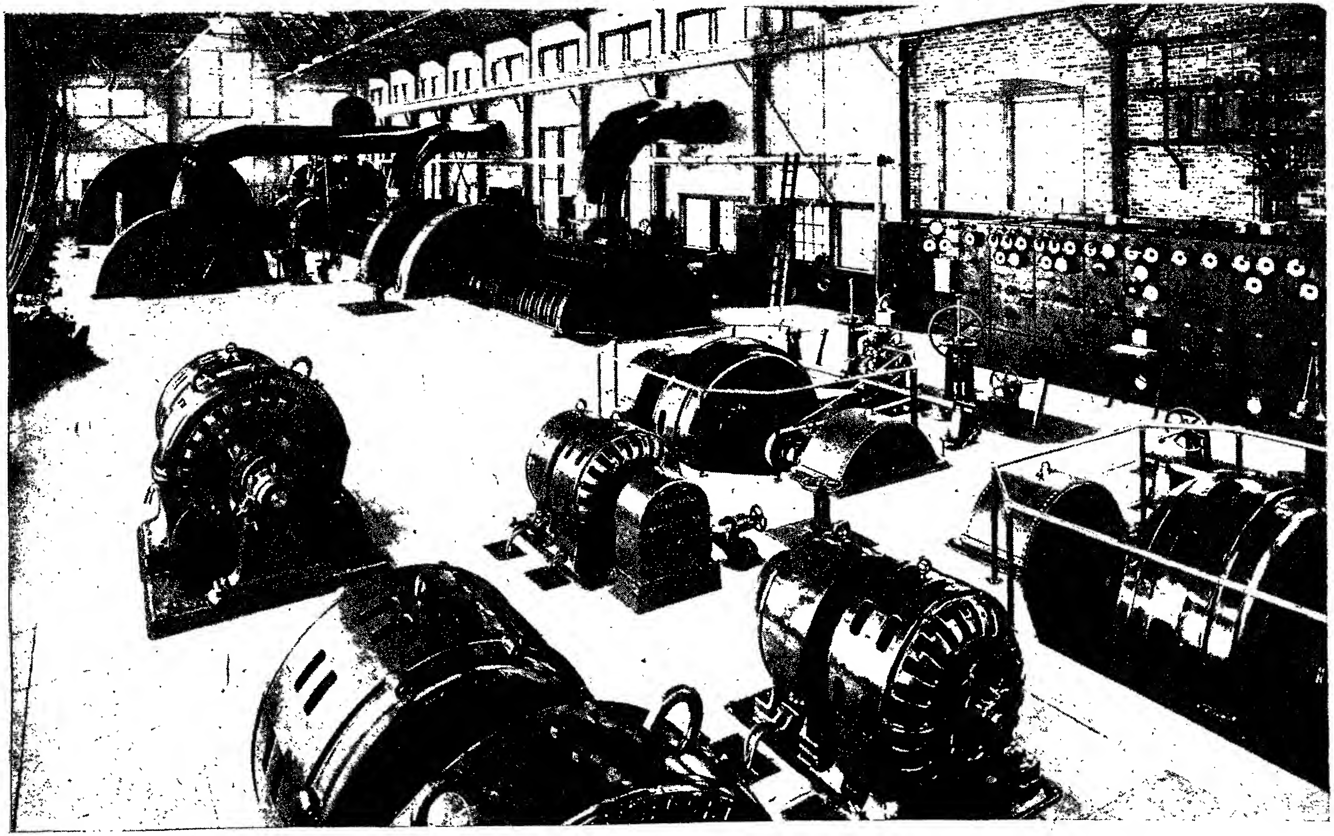


Fig. 51. Power House of Granby Consolidated Mining, Smelting, and Power Company in British Columbia. Generators, Exciters, Blowers, and Compressors Are All Direct-Connected to Pelton-Doble Tangential Units
Courtesy of Pelton Water Wheel Company, San Francisco, California.

The simplicity, cheapness, and high efficiency of this type of water motor commend it for use when the head of water is not less than about 50 feet—though many are in operation with heads of about 25 feet—especially when the supply of water is not abundant. It has the further advantage, due to its simplicity and cheapness, of allowing almost indefinite extension of the existing installation.

In Fig. 49 is shown a typical double "overhung" tangential unit driving an engine type generator. The installation shown in Fig. 50 of four 16,000 h.p. units of the same overhung type is operating under a head of 1400 feet. A very modern installation for the Granby Consolidated Mining, Smelting, and Power Company is shown in Fig. 51. The water is taken from a creek about $\frac{3}{4}$ mile above the power house and furnishes a maximum of 65,000 horsepower. The plant is rather unique in that direct-driving power is provided for running the two main generators, the necessary exciters, two Connorsville blowers, and two Ingersoll compressors, all of them requiring different power and different speeds.

For heads much lower than 50 feet, while this type of motor will, with proper regulation, still give a high efficiency, the construction is such that it cannot utilize a large quantity of water, and therefore the power output will not be great. This disadvantage may be obviated to some extent by mounting several wheels on the same shaft; but in the case of low heads, some form of turbine motor is to be preferred.

In setting up, this wheel must of necessity be placed above the tail-race level, and so high above it that there shall be no danger of interference from back-water. This means that a certain proportion of the total available head must be sacrificed to this condition; and unless the total head is sufficiently great to make the loss thus incurred relatively insignificant, this will not be the best type of motor for obtaining the greatest efficiency from the waterfall (see however, article on "Draft-Tube"). These wheels are well adapted for running high-speed machinery, such as electric generators, air-compressors, etc., by direct connection, thus doing away with much belting or gearing with the attendant loss of power and expense of maintenance. These wheels have been used successfully with heads greater than 2000 feet. They are manufactured in sizes from 6 inches in diameter to more than 30 feet for special cases,

and two or more sizes of nozzle tips are usually provided for adjustment or regulation.

64. Regulation. In connection with the practical working of a water-wheel, an important matter is the quick and efficient control of the discharge from the nozzle in order to vary the power output of the wheel as the load varies, or to conform to fluctuations in the supply of water, so as to maintain a constant speed. Interchangeable nozzles of varying sizes have already been referred to; but this method requires hand manipulation, takes time, and requires attention. When the supply of water is adequate, and the power required sufficiently large, or the load variable, from two to five nozzles may be arranged to play simultaneously around the periphery of the wheel, as shown in Fig. 52. By this means, not only may much greater power be derived from one wheel, but, by shutting off one or more jets, the supply and power may be regulated to correspond to the load fluctuations with very little speed variation. Several wheels may be mounted upon the same shaft, each operated by its own jet or jets; and the regulation or control may be effected by shutting off the supply of one or more wheels, which would then run *dead*. In cases where the supply of water is abundant, so that waste is immaterial, good results can be obtained, especially with the smaller wheels, by mounting the two halves of the vanes on separate wheels (practically dividing the ordinary wheel, with its vanes, into two equal portions by a vertical plane at right

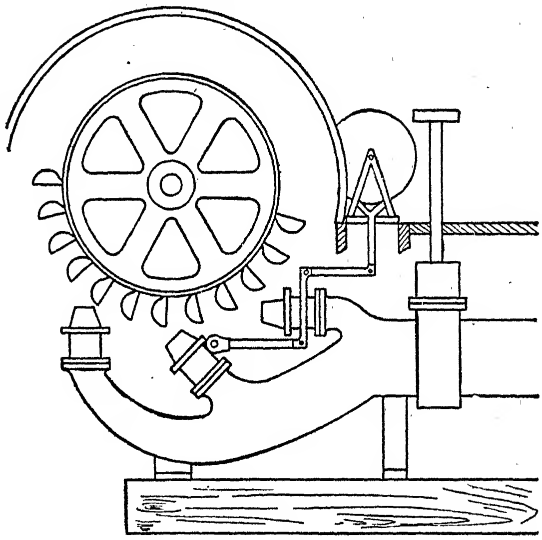


Fig. 52. Wheel Operated by Several Nozzles

angles to the axis). When the wheel is working at full power, the two halves are kept together, and thus form an ordinary wheel of this type; when, however, the speed increases, a governor causes the two wheels

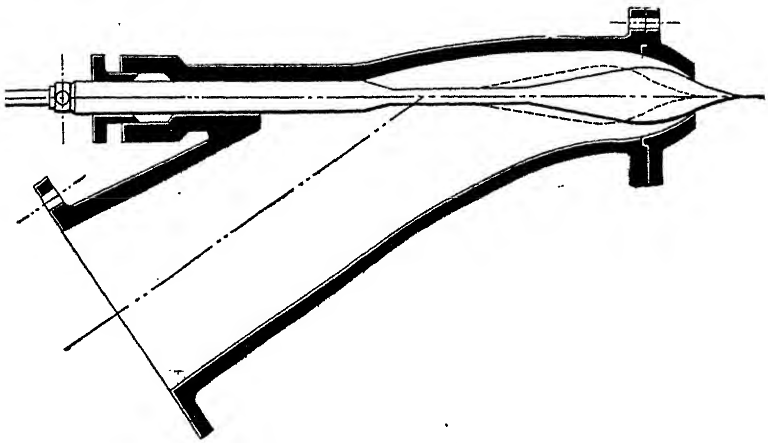


Fig. 53. Section of a Needle Nozzle

to separate more or less, and thus some of the water is allowed to escape between. Several other ingenious devices have been developed for the purpose of accomplishing the same end; a description of some of them, taken mainly from manufacturers' catalogues, follows:

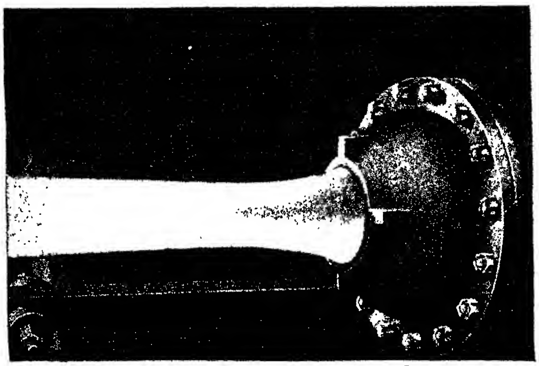


Fig. 54. Stream of Water from Pelton Needle Nozzle Operating under 390-Foot Head and Developing 1,500 H.P.

Note the shadow of needle showing through stream, and the perfect form of jet.

Under average conditions of operation, a governor is not necessary, as, with a constant load, the speed of the wheel is absolutely uniform. When slight and infrequent changes occur—such as are caused by hanging up stamps of a battery, for example—the wheel can be regulated by hand, by means of the main stop-gate, as shown in Fig. 49; but this

would occasion considerable loss of energy, on account of the sudden change of section of the stream. It sometimes happens, however, especially when operating electric plants, that the fluctuations in speed are so sudden and severe that an automatic regulator is essential. In such cases the speed of the wheel may be controlled by various devices, among which may be described the following:

Deflecting Nozzle. The cast-iron *deflecting nozzle* has a ball and socket joint, which permits it to be raised or lowered, thus throwing

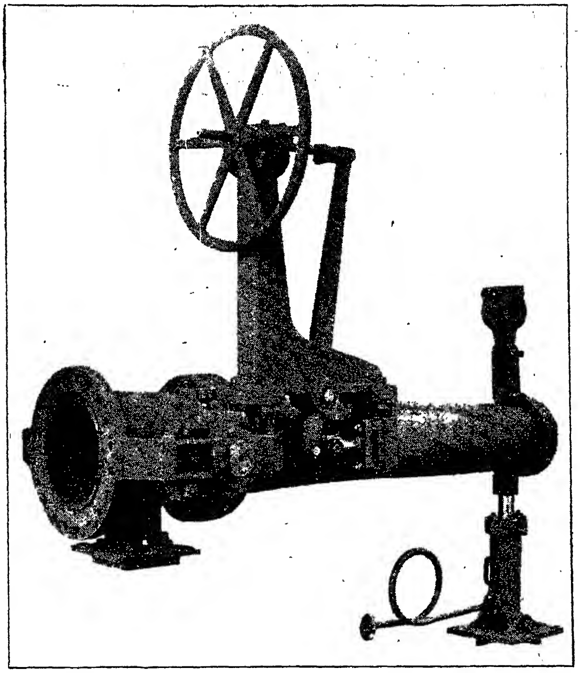


Fig. 55. Pelton-Doble Regulating and Deflecting Nozzle
Courtesy of Pelton Water Wheel Company.

the stream on or off the buckets; the power of the wheel is consequently increased or diminished to correspond to the change of load, and a constant speed is maintained. A steel deflecting plate, which deflects the stream itself—the nozzle remaining stationary—is sometimes used to accomplish the same results when the design will not admit of a deflecting nozzle. Both these devices are wasteful of water; but they effectually prevent *water-hammer,* which would result from a sudden decrease of velocity in the pipe.

Stream Cut-Off. The *stream cut-off*, a spherical plate fitting tightly over the nozzle tip end, by varying its position, changes the discharge area of the nozzle, and thus influences the power of the wheel.

Needle Nozzle. The *nozzle*, Figs. 53 and 54, consists of a body in which is inserted a concentric tapered needle. A change of position of this needle produces a corresponding change of discharge

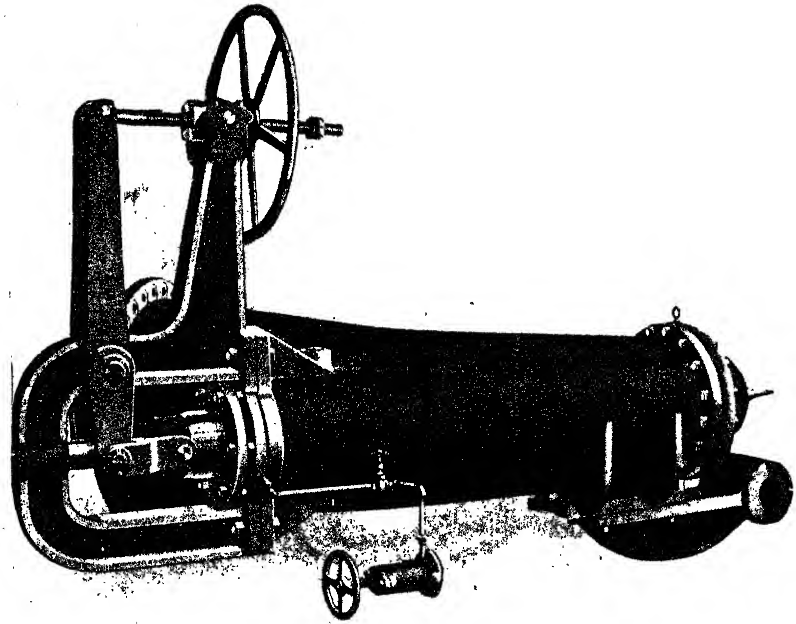


Fig. 56. Pelton-Doble Regulated Needle Nozzle, Arranged for Governor-Controlled Jet Deflectors
Courtesy of Pelton Water Wheel Company, San Francisco, California.

area of the nozzle; the amount of water used is thus varied, and the power of the wheel influenced proportionally.

Needle Regulating and Deflecting Nozzle. A most valuable combination, Fig. 55, consists of a deflecting nozzle swinging on a pair of trunnions, with which is incorporated a needle nozzle, with means for operating either the needle or deflecting nozzle simultaneously or separately. This accomplishes a twofold object—accurate regulation and water economy without water ram. The deflecting nozzle is a most sensitive means of regulation when actuated by an automatic governor, but does not save water. On the other hand, the needle nozzle, while it is extremely economical in the use of

water, is difficult to control quickly by means of the governor. The operation of the combination is as follows:

Assuming the full load to be on the water-wheel, and the nozzle in position of greatest efficiency, a decrease in load, tending to cause increase in speed, will cause the nozzle to be suddenly deflected by the automatic governor. Simultaneously, the needle portion of the nozzle will be actuated by hand, or by another automatic device, tending to close the needle *gradually* and decrease the flow. The governor then raises the nozzle to accommodate the decreased flow of water (and consequent decrease of power), and the nozzle is then brought back to the position of greatest efficiency, having, at the same time, controlled the speed within the required limits.

Another device which, however, is not so economical of water is the hand-regulated needle nozzle with governor-controlled jet

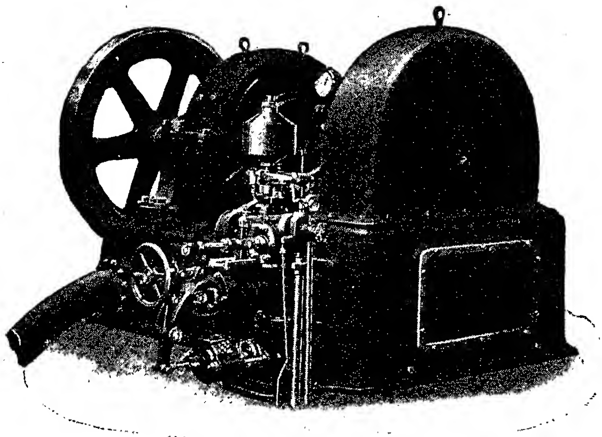


Fig. 57. 100-H. P. Unit Designed for 300-Foot Head, Equipped with Automatic Needle Relief Nozzle and Controlled by Pelton-Doble Governor
Courtesy of Pelton Water Wheel Company, San Francisco, California.

deflector, Fig. 56. A deflector is mounted on the support at the right, and is connected to the governor. A reduction of load would tend to increase the wheel speed, causing the governor to act upon the deflector and reduce the speed by deflecting the jet. Neither of the devices shown in Figs. 55 and 56 give as high an economy of water as the double automatic relief nozzle shown in Fig. 140, Part II.

Governors. In selecting the type of governor to be used, conditions as to head, power, and character of load determine which device or combination is best suited to any individual case.

In Fig. 57 is shown a small 100 h.p. unit, equipped with automatic needle relief nozzle, controlled by a Pelton-Doble governor mounted on the main nozzle casting. The governor is shown separately, Fig. 58. The principle of operation of this governor is somewhat as follows: A fly-ball governor of the Hartun type is enclosed in the revolving cylinder shown at the top. The spindle motion is multiplied by the levers projecting to the right. These levers, actuate a small pilot valve shown at the end of the long spindle projecting through the lever system. This pilot valve permits oil under pressure to either side of a piston valve, which cannot be seen in the figure as it is contained inside the main cylinder casting.

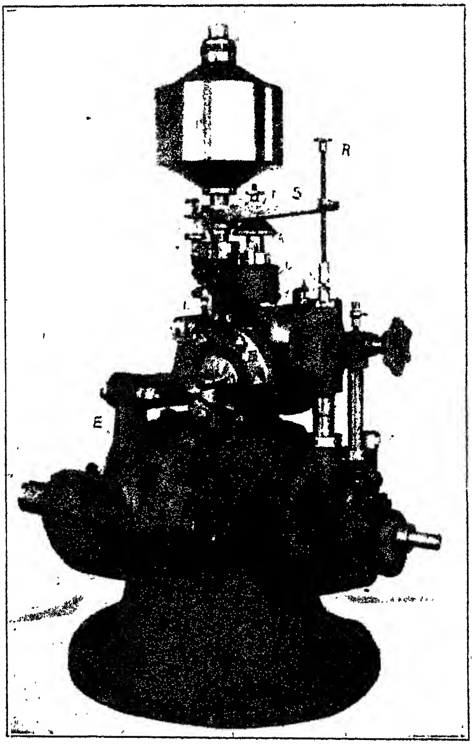


Fig. 58. Modern Type of Pelton-Doble Governor
Courtesy of Pelton Water Wheel Company, San Francisco, California.

This valve in turn permits oil under pressure to enter either end of the main cylinder, the piston rod of which, through suitable link connection, shown in the front of the picture, actuates a rock shaft, which in turn is connected to the needle deflector or the deflecting nozzle, as may be required to meet the conditions of the particular installation.

The long stem projecting through the multiplying levers is an emergency stop device. A slight pressure of the hand downward will quickly bring the entire unit to a standstill.

The governor illustrated is of the self-contained type. In its base it has an oil reservoir with sufficient capacity for four complete

strokes of the governor, and an oil pump of the rotary positive displacement type, driven by the main shaft, maintains the proper pressure on the oil reservoir.

It is readily seen that the ball governor is the ultimate device which actuates or sets in motion the controlling and regulating apparatus. This topic will be further considered under "Turbines" (See Part II).

In the case of long pressure-pipes, especially when under high pressure, it is difficult and dangerous suddenly to vary the quantity

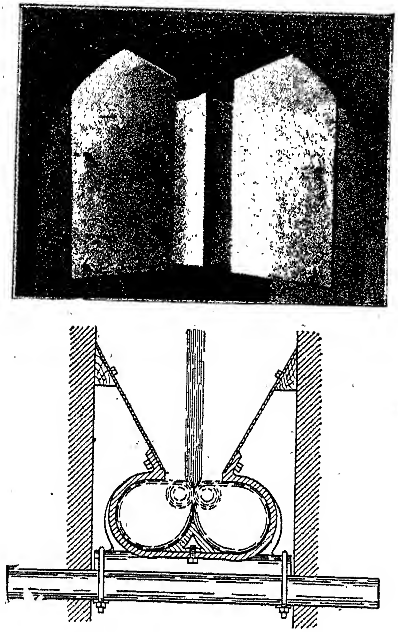


Fig. 59. "Ensign" Vortex Baffle-Plate as Installed in a Tail-Race

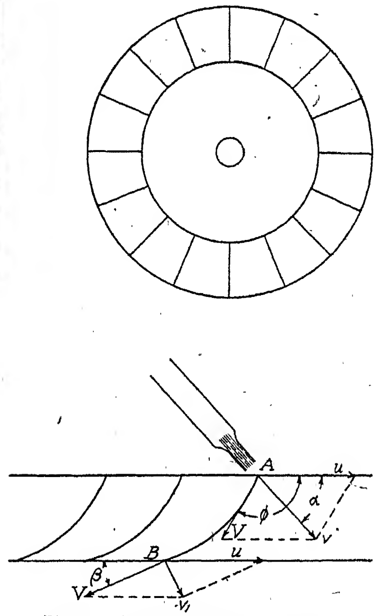


Fig. 60. Downward-Flow Impulse Wheel

of water delivered by the nozzle in such a manner as is necessary to regulate the speed of a hydro-electric generating unit subject to sudden violent variations of load. Consequently it has become customary to regulate the speed of such units by deflecting the jet of water, so that all, or part of it, misses the water-wheel buckets, and is for the moment necessarily wasted. The water which is thus prevented from giving its energy to the water-wheel, is projected through the tail-race at a very high velocity—in some cases exceeding 300 feet per second (18,000 feet per minute)—and becomes destruc-

tive, particularly when the water unavoidably carries infinitesimal particles of sand. No masonry can long withstand the action of such a jet, and even iron and steel are rapidly worn away, as if by a terrific sand-blast.

Baffle-Plate. The *Ensign Vortex Baffle-Plate* (patented), Fig. 59, is designed to divide such a jet in halves, and deflect the halves until they impinge upon each other, and harmlessly spend their force. The device is a trough-like structure with a sharp central vertical dividing wedge, made to be replaceable in case of wear. The device is intended to split the impinging jet, so that it reacts on itself and falls into the tail-race with comparatively little disturbance. In practice such results are not always entirely realized.

SPECIAL JET TYPES

65. Downward-Flow Impulse Wheels. In this type of motor, the horizontal impulse wheel is driven by the jet from a nozzle inclined downward at a convenient angle, as in Fig. 60, which represents in outline the plan and the development of part of a cylindrical section of such a wheel. The water, in passing through the wheel, neither approaches nor recedes from the axis of rotation; it is therefore sometimes called a *parallel-flow* or *axial-flow* wheel.

The stream enters at A, as shown, with the relative velocity V; passes downward over the vane, always maintaining the same distance from the axis; and, *neglecting the effect of friction and gravity,* issues from the vane at B, with the same relative velocity V.

As before, to prevent impact losses at A, the direction of the relative velocity V must be tangent to the vane at that point; and in order that the efficiency should be high, the absolute velocity of departure v_1 must be small, which latter condition will be fulfilled if $u = V$ at B. Therefore, as explained in the preceding analyses, ϕ should be made equal to 2α, and the best speed of the wheel is $u = \dfrac{v}{2 \cos \alpha}$. The efficiency under these conditions is:

$$e = 1 - \left(\frac{\sin \frac{1}{2} \beta}{\cos \alpha}\right)^2,$$

which again shows that both α and β, particularly the latter, should be small for high efficiency.

In the above analysis, no account was taken of the force of

gravity acting as the water descends through the vertical distance between A and B; this would increase the efficiency and the advantageous velocity above the values as found from the equations above.

It is evident that several nozzles might be employed also with this type of wheel, instead of one, where the supply of water is adequate.

(Articles 11 and 12 develop the hydraulic formulae to be used in problems of nozzle discharge. Article 14 shows the proper relation between the diameters of nozzle and pipe to furnish maximum power; and Article 16 considers the case of multiple nozzles to fulfill the same condition.)

66. Girard Impulse Wheels. This type of wheel (Fig. 61) consists essentially of two flat, parallel, and concentric rings or *crowns*, between which are inserted the curved vanes or blades, the whole attached rigidly to the axle and forming the wheel proper, or *runner*. The feed or operating water issues from a nozzle placed inside the wheel as shown, in which case it is an *outward-flow* impulse wheel; or the nozzle may be placed outside, making it an *inward-flow* wheel; or several nozzles or groups of nozzles may be employed, located symmetrically around the circumference. The analyses and conclusions contained in the preceding articles apply to these cases.

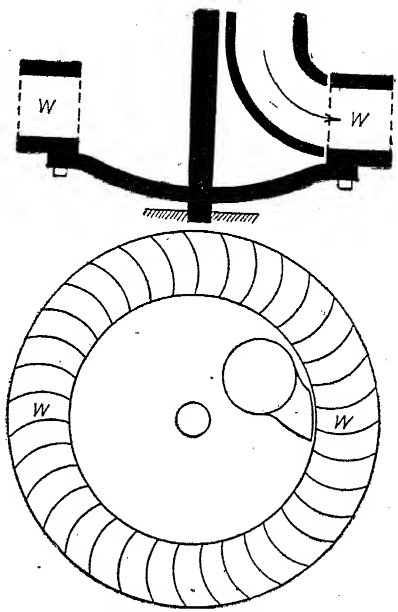


Fig. 61. Girard Impulse Wheel

Axial or parallel flow may be applied to this type of wheel, as explained in Article 65. Occasionally, with the outward-flow type, the two crowns are diverged to avoid choking the passageways.

Openings in the crowns to facilitate the escape of air are frequently made with the same object in view. This type of wheel has been widely used in Europe, a number of single motors of this kind developing 1,000 horse-power.

In the electric power station at Vernayaz, Switzerland, are six 1,000-horse-power Girard wheels, working under a head of 1,640 feet; the outer diameter of each wheel is about 6.5 feet; and the normal speed, about 540 revolutions per minute. These wheels work with but one *guide* (the nozzle tube) each.

A turbine built for the Ouiatchouan Pulp Company (Quebec) has two sets of such guides spaced 180 degrees apart. This wheel develops 1,000 horse-power under a head of 240 feet, running at 225 revolutions per minute; it is enclosed in a cast-iron case and provided with a draft-tube and air-admission valve, both of which contrivances will be described in a later article.

Example. Let us assume a Girard outward-flow impulse wheel, with $\alpha = 25$ degrees; $\beta = 20$ degrees; ratio $\frac{r_1}{r} = \frac{4}{3}$; supplied with 2 cubic feet per second through 12-inch pipe 2,000 feet long, with nozzle attached, having a coefficient of velocity of 0.95. Total head over nozzle tip, 152.00 feet, of which 8.3 feet are consumed in pipe friction and entrance losses. Wheel to make 240 r.p.m.

Velocity in the pipe is:

$$v_p = \frac{q}{a} = \frac{2}{\pi \frac{1}{4}} = 2.55 \text{ feet per second.}$$

Velocity of jet is:

$$v = 0.95 \sqrt{2g\,(152.00 - 8.3) + 2.55^2} = 91.4 \text{ feet per second.}$$

The best speed for the inner rim is

$$u = \frac{v}{2 \cos \alpha} = \frac{91.4}{2 \times 0.906} = 50.5 \text{ feet per second}$$

Since $2\pi r n = 50.5$.

$$r = \frac{50.5}{2\pi \frac{240}{60}} = 2.01 \text{ feet.}$$

$$r_1 = \frac{4}{3} \times 2.01 = 2.68 \text{ feet.}$$

The theoretic efficiency is:

$$e = 1 - \left(\frac{r_1}{r} \frac{\sin \frac{1}{2} \beta}{\cos \alpha}\right)^2 = 1 - \left(\frac{4}{3} \frac{0.174}{0.906}\right)^2 = 0.93.$$

The actual efficiency would probably have a value between 75 and 80 per cent.

The work imparted to the wheel is, theoretically:

Work $= W\dfrac{v^2}{2g} = 2 \times 62.5 \times \dfrac{\overline{91.4}^2}{64.4} = 16{,}213$ ft.-lbs. per second, if the nozzle be not considered a part of the motor, and if losses in the wheel itself be disregarded.

If the nozzle be considered part of the motor, the work imparted to it, disregarding wheel losses, is:

$$\text{Work} = wq\left(152.00 - 8.3 + \frac{\overline{2.55}^2}{2g}\right)$$

$$= 62.5 \times 2\left(143.7 + \frac{6.5}{64.4}\right) = 17{,}975 \text{ ft.-lbs. per second.}$$

If the wheel, under the second assumption, have an efficiency of 75 per cent, the useful work of the wheel is:

Useful work $= 17{,}975 \times 0.75 = 13{,}475$ ft.-lbs. per second;

$$\frac{13{,}475}{550} = 24.5 \text{ horse-power.}$$

REACTION WHEELS

67. Rotating Vessels. As a preliminary to the study of the theory and operation of turbines, it will be necessary to consider very briefly some of the essential features of the action of rotating fluids.

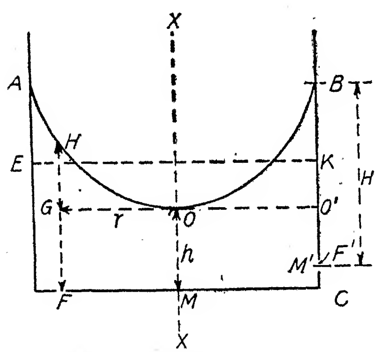


Fig.62. Revolving Open Vessel.

Flow from a Revolving Vessel with Free Surface. Let AC (Fig. 62) be any open vessel, revolving about a vertical axis XX. It is shown in treatises on Hydraulics, that the water surface EK, which is horizontal before rotation, becomes, under the action of centrifugal force and gravity, a curved surface AOB, due to the rotation; that this surface is a paraboloid of revolution, so that any vertical section through the axis is a parabola with the vertex at O, and axis vertical and coincident with the axis of rotation; that the head of water over an orifice F in the base or side, at any distance r from the axis of rotation, is $h + \dfrac{u^2}{2g}$, if u be

linear velocity of the center of the orifice, and h its distance below O; and, therefore, that the relative velocity of efflux from F is:

$$V = \sqrt{2g\left(h + \frac{u^2}{2g}\right)} = \sqrt{2gh + u^2} \quad . \; . \; . \; . \; . \; (82)$$

Let n be the number of revolutions per second; then $u = 2\pi rn$; and

$$V = \sqrt{2gh + 4\pi^2 r^2 n^2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (83)$$

This result is independent of the shape of the containing vessel; and the axis of rotation may lie within or without it, the axis of the paraboloid in any case coinciding with the axis of rotation.

68. Closed Vessel. The above formulæ apply equally well to the case of a closed rotating vessel in which the curved surface is wholly or partially prevented from forming, as in Fig. 63. Here also h is the depth MO in the axis of rotation; and the parabola AOB represents the vertical section of the paraboloid of *pressures*. In both cases, then,

$$\frac{u^2}{2g} = \frac{2\pi^2 r^2 n^2}{g}$$

is the head GH, to be added to the minimum static head MO at the axis, to obtain the total pressure head over the orifice. If the orifice is in the vertical bounding wall of the vessel, as at F', the pressure head is $M'O' + O'B = H$.

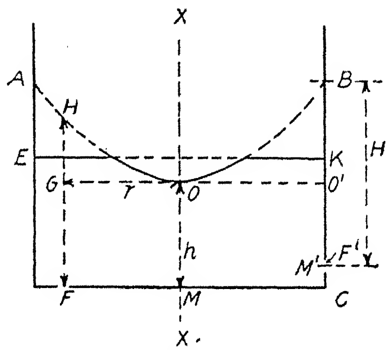


Fig. 63. Revolving Closed Vessel.

69. Revolving Tubes. Fig. 64 represents the simple case of one or more hollow arms attached to a vessel, and rotating with it about a vertical axis. From what has preceded, it is clear that the static pressures at the points A and B in the tube, when rotation has been established, but *when no flow occurs*, are, respectively:

$$OM + GH = h + \frac{u^2}{2g}, \text{ for the point } A\text{; and,}$$

$$OM + G_1H_1 = h + \frac{u_1^2}{2g} \text{ for the point } B,$$

if u and u_1 be the linear velocities of the points A and B respectively.

When the orifices are opened and *flow takes place*, the pressure-head in each case falls by an amount equal to the velocity head *plus*

the head lost in frictional resistances, as explained in Articles 5 and 6; and the line of pressure now assumes some other form, such as LB. Neglecting for the present the frictional losses, it is evident that the following relations must obtain, by reason of the principle of the conservation of energy:

$$h + \frac{u^2}{2g} = h' + \frac{V^2}{2g},$$

which becomes, for the point B, since no pressure head exists at the end of the tube when it discharges freely into the air:

$$h + \frac{u_1^2}{2g} = O + \frac{V_1^2}{2g}, \quad \ldots\ldots\ldots\ldots \quad \textbf{(83a)}$$

so that

$$h = h' + \frac{V^2}{2g} - \frac{u^2}{2g} = \frac{V_1^2}{2g} - \frac{u_1^2}{2g} \ldots \quad \textbf{(83b)}$$

if V and V_1 represent the relative velocities in the tube at the points A and B, respectively. If the tube is submerged, there is static pressure at the end; so that, if h'' is the static pressure on the end (the depth of submergence), then,

$$h + \frac{u_1^2}{2g} = h'' + \frac{V_1^2}{2g}, \quad \ldots\ldots\ldots\ldots \quad \textbf{(84)}$$

and therefore,

$$h = h' + \frac{V^2}{2g} - \frac{u^2}{2g} = h'' + \frac{V_1^2}{2g} - \frac{u_1^2}{2g}. \quad \textbf{(84a)}$$

The above equation (84a) expresses the relation between the pressure-head, velocity-head, and rotation-head at any point of a revolving tube. In case the tube is only partly full, as when a stream impinges and glides along a vane (or one side of a tube or bucket of a water-motor), there can be no static pressure, and the above becomes:

$$V_1^2 - V^2 = u_1^2 - u^2, \quad \ldots\ldots\ldots\ldots \quad \textbf{(84b)}$$

which is Equation 51, for the case of a jet impinging on a vane.

Work of a Reaction Wheel. Fig. 64 represents essentially a reaction wheel, since the dynamic pressure causing rotation is caused entirely by the reaction of the issuing jets. In order to discuss the work and energy of such an apparatus, we may use Equation 57, which expresses the work of the impulse of the entering stream and the reaction of the departing stream, by simply omitting the term representing the former. Accordingly, for the work of a reaction wheel:

$$\text{Work} = W\frac{u_1 V_1 \cos\theta}{g} \quad \dots\dots\dots\dots\dots\dots \quad (85)$$

$$= W\frac{u_1 V_1 \cos\beta - u_1^2}{gh} \text{(from Equation 55)} \quad \dots\dots \quad (86)$$

$$= W\frac{u_1 \cos\beta \sqrt{2gh + u_1^2} - u_1^2}{g} \text{(from Equation 83a)} \cdot (87)$$

Dividing the expression for Work by the theoretic energy Wh, we have:

$$\text{Efficiency} = \frac{u_1 \cos\beta \sqrt{2gh + u_1^2} - u_1^2}{gh} \quad \dots\dots\dots\dots \quad (88)$$

The work is zero when $u_1 = 0$—that is, when there is no rotation; also when $u_1^2 = 2gh \cdot \cot^2\beta$; and it is a maximum, and equal to

$$\text{Work (Max.)} = Wh\,(1 - \sin\beta) \quad \dots\dots\dots\dots\dots\dots \quad (89)$$

when,

$$u_1^2 = \frac{gh}{\sin\beta} - gh, \quad \dots\dots (90)$$

the efficiency, in this case, being:

$$e\ (\text{Max.}) = 1 - \sin\beta \quad \dots\dots (91)$$

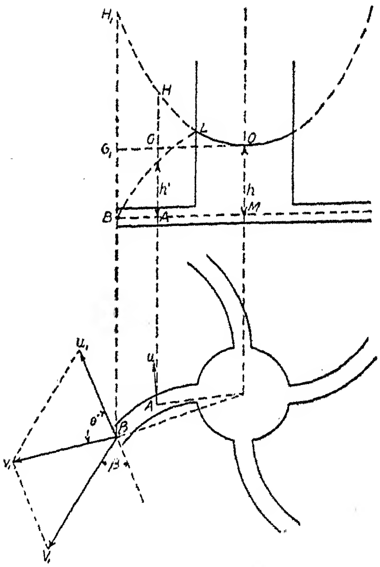


Fig. 64. Revolving Vessel with Hollow Arms Attached.

The work and efficiency, therefore, increase as the angle β decreases. When $\beta = 90$ degrees, the work and the efficiency both become zero, for the jet in such case issues radially; when $\beta = 0$ degrees, the work is Wh, and the efficiency is unity, or 100 per cent; but the velocity u_1 (*and therefore also* V_1) becomes infinitely great. It must be remembered that frictional and air resistances have not been considered in the above analysis; both increase rapidly with increased speed of rotation. In general, however, it may be stated that within certain limits the efficiency of a reaction wheel increases with the speed and with the smallness of the angle β; and it is greatest in any given case, when the angle β is zero—that is, when the water issues in a direction exactly opposite to that of rotation.

70. Barker's Mill. Fig. 65 represents an apparatus com-

monly known as *Barker's Mill.* It is the reaction wheel described in the preceding article, with the direction of the issuing streams of water directly opposite to that of revolution, or $\beta = 0$. Making $\beta = 0$ in the preceding equations, we have:

$$\text{Work} = W \frac{u_1 \sqrt{2gh + u_1^2} - u_1^2}{g}; \quad \ldots\ldots (92)$$

$$\text{Efficiency} = \frac{u_1 \sqrt{2gh + u_1^2} - u_1^2}{gh}; \quad \ldots\ldots (93)$$

$$\text{Work (Max.)} = Wh; \quad \ldots\ldots (94)$$

$$\text{Efficiency (Max.)} = \text{unity, or 100 per cent,} \quad \ldots\ldots (95)$$

when u_1 = infinity; in which case also v_1 = infinity.

If a_1 be the area of the exit orifices, and w the weight of a cubic unit of water, the weight of water discharged in one second is wa_1v, which becomes infinite when $u_1 = V_1$ = infinity. As stated before, frictional and air resistances increase rapidly with the speed, so that the above relations, in deriving which these resistances have not been considered, are theoretic. It is evident, however, that the efficiency of a reaction wheel of this type increases with the speed within certain limits; and that the discharge varies with the speed.

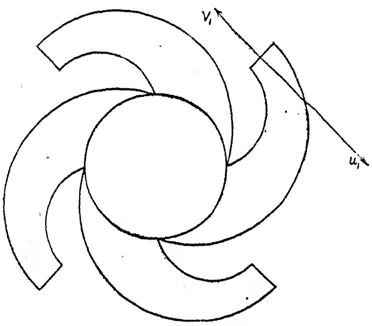


Fig. 65. Barker's Mill.

71. Effect of Friction. If c_v be the coefficient of velocity representing the effect of friction in the arms and orifice, then,

$$V_1 = c_v \sqrt{2gh + u_1^2}, \quad \ldots\ldots (96)$$

instead of the theoretical expression,

$$V_1 = \sqrt{2gh + u_1^2}$$

The expressions for the effective work of the wheel and the efficiency then become:

$$\text{Work} = W \frac{c_v u_1 \sqrt{2gh + u_1^2} - u_1^2}{g} \quad \ldots (97)$$

$$\text{Efficiency} = \frac{c_v u_1 \sqrt{2gh + u_1^2} - u_1^2}{gh}, \quad \ldots (98)$$

$$\text{Efficiency (Max.)} = 1 - \sqrt{1 - c_v^2} \quad \ldots\ldots (99)$$

when,

$$u_1^2 = \frac{gh}{\sqrt{1 - c_v^2}} - gh \cdots\cdots \quad \textbf{(100)}$$

If $c_v = 1$—that is, when frictional loss is not considered—$e = 1$; and $u_1 = V_1$ = infinity, as before. When $c_v = .94$, the advantageous velocity $u_1 = \sqrt{2gh}$, and the efficiency is 65 per cent. Thus the effect of friction is greatly to decrease the theoretic efficiency. To render c_v large, the tubes should be smooth and well rounded by means of easy curves. In addition to the above considerations, the air resistance, which has not been included in the above analysis, increases very rapidly with the speed of rotation, and its effect is to reduce still further the computed efficiency. Because of the low actual efficiency resulting from the above factors, the reaction wheel is not used as a hydraulic motor.

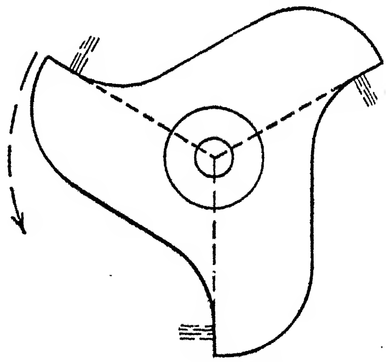


Fig. 66. Scotch Wheel

72. Improvements in Design. *Scotch Wheel.* The Scotch wheel (Fig. 66) is an improvement on the Barker's Mill; the three orifices are made adjustable in size by means of movable flaps, for the purpose of regulating the quantity of water and the power.

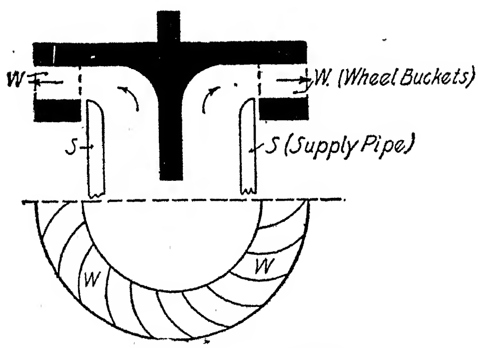


Fig. 67. Combe's Turbine

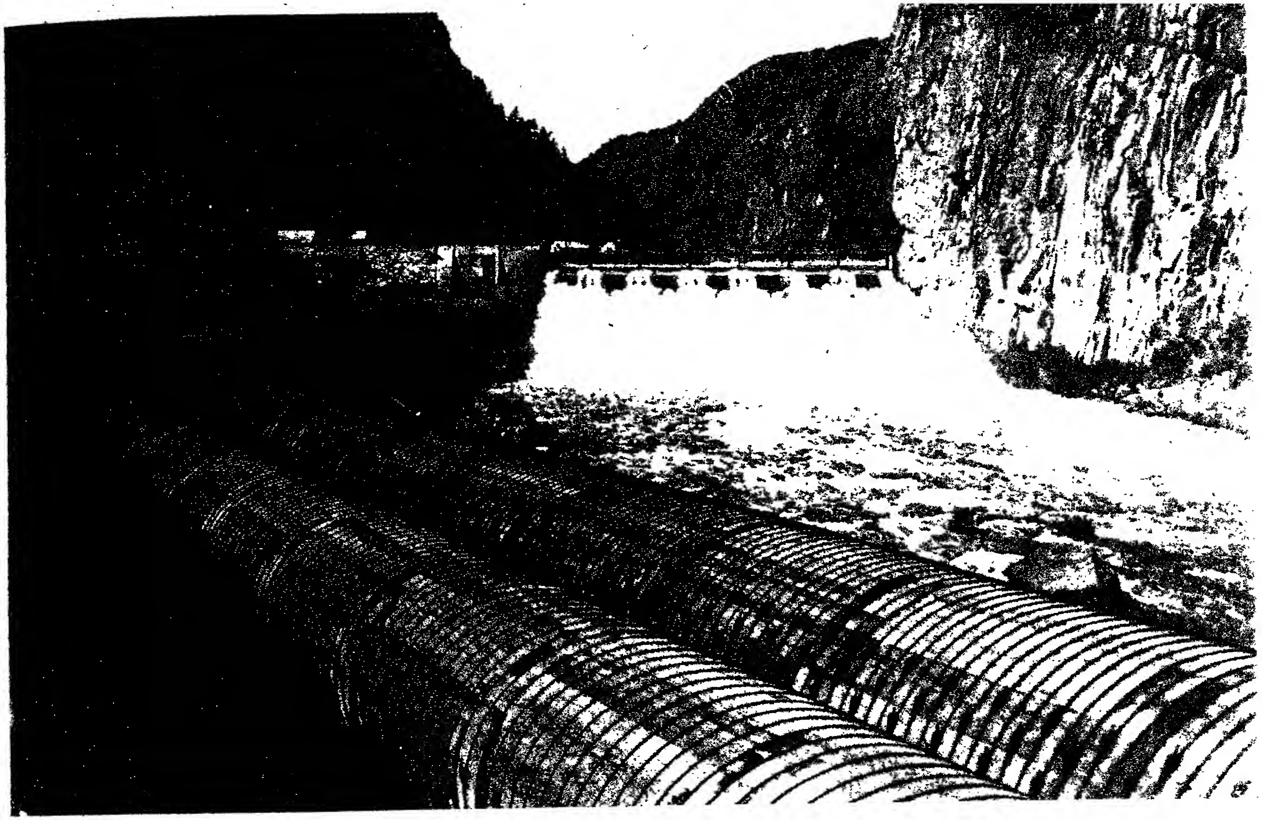
Combe's Turbine. An increase in the number of discharging streams, as in Combe's turbine, [Fig. 67, constituted the next advance in design. The supply pipe conducts the water directly to the runner buckets, there being no guide vanes. The passageways are completely filled with water which discharges into the air with a comparatively low absolute velocity.

The *Cadiat turbine*, a modification of Combe's type, has the supply pipe above it.

73. Modern Turbine Development. The next noteworthy advance in design and construction was due to the French engineer Fourneyron, who, in 1826, improved the Cadiat turbine by using stationary guide blades within the entire wheel ring to impel the entering water in a forward direction. This resulted in a low absolute velocity of the discharge water, and, the wheel being entirely under water, the wheel channels were completely filled. This type of wheel is usually considered the first modern turbine, which has since been considerably improved and varied in Europe and America, until, at the present time, it has practically superseded the older forms of water wheels.

By the middle of the nineteenth century Fourneyron had developed the radial outward-flow turbine known under his name. Mr. Boyden improved upon this type and introduced into America Fourneyron's diffuser, which is therefore frequently called the *Boyden diffuser*, a form of apparatus now practically obsolete, its place being taken by the *draft tube*, but the principles underlying their action being the same in both cases. Mr. Francis built a radial inward-flow or vortex turbine, which, under a head of 19 feet showed an efficiency of practically 80 per cent.

The speed of revolution of a turbine varies directly as the square root of the head of water and inversely as the diameter. Consequently, as low heads were the rule on the early turbines, it was soon found necessary to reduce the wheel diameters in order to meet the demand for greater speed of shafting. This reduction in diameter and, in consequence, of the interior space compelled the manufacturers to increase the bucket in depth, that is to say in an axial direction, at the same time changing the curvature of the buckets from a radial to a more or less axial direction. This tendency was further strengthened by the consideration that any decrease in diameter of wheel without a corresponding increase in depth would result in a reduction of the entrance and exit areas, and interfere with the proper motion of the water through the wheel, thus lowering the efficiency. As a consequence of these developments, depth of bucket, great capacity in proportion to diameter, high speed, and peculiar form of bucket, are characteristic of the *American* turbines.



MADISON DAM, ROCK FILL TIMBER CREEK AND PIPE LINES

The dam is 44 feet high and 183 feet long. The wood stave pipes are 10 and 12 feet in diameter, respectively, and are 7500 feet long.

Courtesy of Montana Power Company

WATER-POWER DEVELOPMENT

PART II

TURBINES AND WATER WHEELS

GENERAL TYPES

74. Elements. A turbine consists essentially of a series of short, curved passageways or buckets divided from one another by vanes or blades, the whole forming a single rigid body attached to the axle, and called the *runner*. The water for operating the runner passes into the passageways through a set of fixed or stationary channels called *guides*, the feed water being admitted around the entire circumference. For convenience and efficiency of operation, the turbine is provided with various controlling, regulating, and governing devices (Fig. 68).

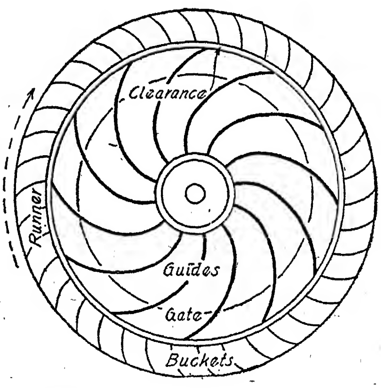


Fig. 68. Diagram of Typical Turbine

75. Classification. *Direction of Flow.* Turbines are classified in several ways, depending on the basis used. Thus, with respect to the direction of flow of the water through the wheel, there are three classes—*radial-flow, axial-flow,* and *mixed-flow* turbines.

A *radial-flow* turbine is one in which the path of a particle of water within the wheel lies in a plane perpendicular to the axis of rotation. The direction of flow may be either outward or inward; that is, the turbine may have internal or external feed (Figs. 69 and 70). An *axial-* or *parallel-flow* turbine is one in which the distance of a particle from the axis of rotation remains constant during its passage through the wheel (Fig. 71). *Mixed flow* is a combination of radial and axial flow; it is usually inward and axial.

Pressure Conditions. Another classification is *impulse* and *reaction* turbines. If the wheel passages are not completely filled with water, and if air enters freely so that the entire stream within each wheel passage is under atmospheric pressure, the motor is called an *impulse* turbine. If the wheel passages are completely filled by the water flowing through them under pressure, the motor is called a *reaction* turbine.

The *limit* turbine is an intermediate form possessing characteristics common to both the

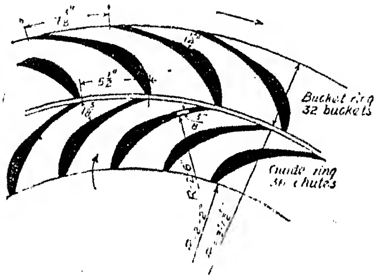


Fig. 69. Section of Guides and Buckets, Fourneyron Turbine, Niagara Falls

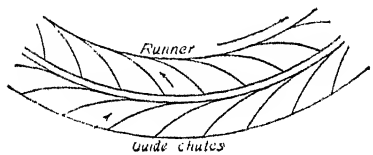


Fig. 70. Section of Runner of Francis Center-Vent Turbine

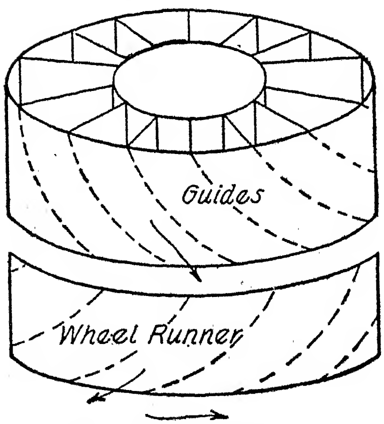


Fig. 71. Axial-Flow or Parallel-Flow Turbine

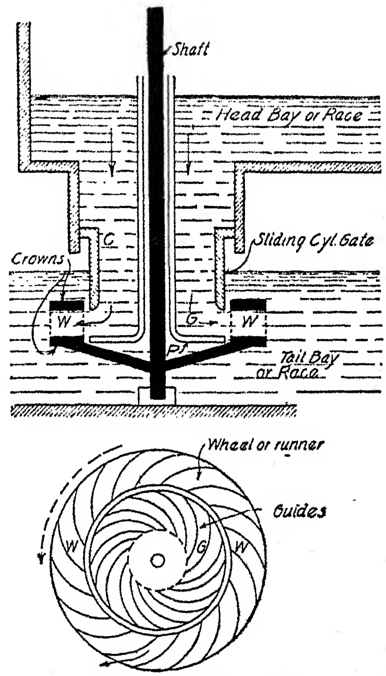
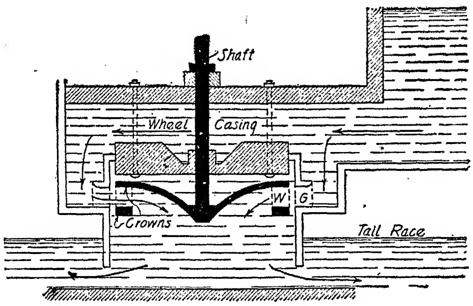


Fig. 72. Fourneyron Radial Outward-Flow Turbine

impulse and reaction types. Though it has many promising features, it has not as yet engaged the attention of designers to any extent.

Inventors. The three typical classes above described are often called by the names of the eminent hydraulicians who invented

or perfected them. Thus, the reaction turbine with radial outward flow is frequently called the *Fourneyron* turbine, Fig. 72. The guides *G* are rigidly attached to the fixed plate *P*, which is connected with the hollow pipe enclosing the shaft. In such a wheel the discharge may be either into the air or into a body of water. A suction or draft tube cannot very conveniently be used with this type of motor.



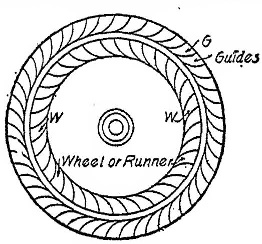


Fig. 73. Francis Radial Inward-Flow Turbine

A reaction wheel similar to the above, but with radial inward flow, is often called a *Francis* turbine, Fig. 73. A reaction turbine with axial flow is generally termed a *Jonval* turbine, Fig. 74. The discharge from the two latter types of motor may take place into the air, directly into the tailwater, or into a suction (draft) tube.

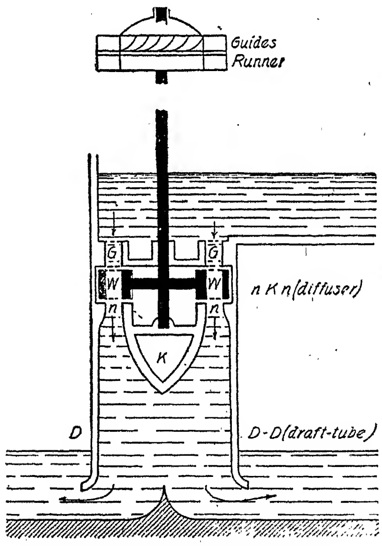


Fig. 74. Jonval Axial-Flow Turbine

Limit turbines are sometimes called *Haenel* turbines; they may be considered impulse turbines without free deviation.

The most common forms of reaction turbines used in America, particularly for the smaller sizes, are of the mixed-flow type, having radial inward admission and axial downward discharge, as the *Swain* turbine; or of the plain inward-flow type.

Turbines of the "American" pattern in which the water passes radially inward, then axially, and finally leaves the buckets between the axial and the radial outflow direction, are called *American vortex* turbines, types of which will appear in subsequent articles.

76. Usage. Any turbine may be made to act as either an impulse or a reaction turbine. If the conditions are such that the water, in passing through the vanes, fails to fill them completely, it is an impulse turbine; if the wheel be placed under water, "drowned", or if by any other means the water is compelled to completely fill all the passages under pressure, it acts as a reaction turbine.

*"The type to be employed in each individual case should be in accordance with the height of the head to be utilized, as follows:

(1) For *low* heads, up to 40 feet: American type of turbine, i.e., of the 'inward and downward' variety, with horizontal or vertical shaft in open flume or case, nearly always with draft tube.

(2) For *medium* heads, from 40 to 300 or 400 feet: Radial inward-flow reaction or *Francis* turbine, with horizontal shaft and concentric or spiral cast-iron case with draft tube.

(3) For *high* heads, above 300 or 400 feet: Impulse wheel of the usual type, as the Pelton and similar wheels; or radial outward-flow segmental-feed, free deviation, as the Girard impulse wheel; or a combination of both, with horizontal shaft and cast- or wrought-iron case, often with draft tube."

Recent improvements in design, however, have made it possible to employ the reaction turbine in connection with heads considerably higher than stated above; for example, the Noriega development in Mexico consists of 6000-horsepower single-runner Francis turbines operating under a head of 670 feet, at a speed of 514 revolutions per minute.

ESTIMATES FOR WATER POWER†

77. Consideration of Methods. The methods of estimating the water power that can be derived by damming a stream are similar in the main features to those for water supply and will not be treated

* John Wolf Thurso, "Modern Turbine Practice", D. Van Nostrand Co., New York.

† Articles 77 to 81 inclusive have been taken, with slight changes, from Prof. Mansfield Merriman's "Treatise on Hydraulics", John Wiley & Sons, New York.

here in detail. In the absence of gagings, the records of rainfall and evaporation are to be collected and discussed; but a few gagings will give much more definite information, if records of water stages during several years can be had. Here, also, the minimum flow of the stream must receive careful attention, particularly when the plant is to generate electrical power for trolley and light service, for the interruption of such service is a serious public inconvenience.

In nearly every situation the stream flow in connection with the storage which can be obtained at a reasonable expense is not sufficient to continuously generate the power which is required. In such cases it is necessary to supplement the water power with an auxiliary steam plant located at some point within the territory to be served, where fuel can be obtained most economically. In order to determine the capacity of the auxiliary plant, as well as the advisable extent of the water-power development from the available records of stream flow, the method developed by Mr. Clemens Herschel* may be employed.

78. Efficiencies. *Motor.* Let W be the weight of water delivered per second to a hydraulic motor, and h be its effective head as it enters the motor, h being due either to pressure or to velocity, or to pressure and velocity combined. The theoretical energy per second K of this water equals Wh. If W be in pounds, and h in feet, the theoretical horsepower, HP., of the water as it enters the motor is $Wh \div 550$; and this is the power that can be developed by a motor of efficiency unity. The work k delivered by the motor is, however, always less than K, owing to losses in impact and friction, and the horsepower hp. of the motor is less than HP. The efficiency of the motor is

$$e=\frac{k}{K}=\frac{k}{Wh} \qquad \text{or, } e=\frac{\text{hp.}}{\text{HP.}}$$

and the value for turbine wheels is usually taken as 0.75 to 0.80; that is, the wheel transforms into useful work about 75 per cent to 80 per cent of the energy of the water that enters it. In recent designs, however, 90 per cent and higher values have been realized.

Approaches. In designing a water-power plant, it should be the aim so to arrange the forebays and penstocks which lead the

* Transactions of American Society of Civil Engineers, 1907, Volume 58.

water to the wheel that the losses in these approaches may be as small as possible. The entrance from the headrace into the forebay, from the forebay into the penstock, and from the penstock to the motor, should be smooth and well rounded; sudden changes in cross-section should be avoided; and all velocities should be low, except that at the motor. If these precautions are carefully observed, the loss of head outside the motor can be made very small.

Let H be the total head from the water level in the headrace to that in the tailrace below the motor. The total available energy per second is WH; and it should be the aim of the designer to render the losses of head in the approaches as small as possible, so that the effective head h may be as nearly equal to H as possible. Neglect of these precautions may render the effective power less than that estimated.

The efficiency e_1 of the approaches is the ratio of the energy K of the water as it enters the wheel, to the maximum available energy WH, or

$$e_1 = \frac{K}{WH}$$

Plant. The efficiency E of the entire plant, consisting of both approaches and wheel, is the ratio of the work k delivered by the wheel to the energy WH; or

$$E = \frac{k}{WH} = \frac{eK}{WH} = ee_1$$

or, the final efficiency is the product of the separate efficiencies. If the efficiency of the wheel be 0.75, and that of the approaches 0.96, the efficiency of the plant as a whole is 0.72, or only 72 per cent of the theoretical energy is utilized. Usually the efficiency of the approaches can be made higher than 96 per cent.

In making estimates for a proposed plant, the efficiency of turbine wheels may be taken at 80 per cent; the effective work is then 0.80 Wh, where h is the actual effective head on the motor; and accordingly, if the wheels are required to deliver the work k per second, the approaches are to be arranged so that Wh shall not be less than $1.25k$. Especially when the water supply is limited, is it important to make all efficiencies as high as possible.

79. Water Delivered to Motor. To determine the efficiency of a hydraulic motor by formula, k is to be measured by the methods

of Article 81, and h found by Article 80. In order to find the weight W that passes through the wheel in one second, there must be known the discharge per second q, and the weight w of a cubic unit of water; then

$$W = wq$$

Here w may be found by weighing one cubic foot of the water; or, in approximate computations, w may be taken at 62.5 pounds per cubic foot. In precise tests of motors, however, its actual value should be ascertained as closely as possible.

Flow Measurement. Measurement of flow through orifices, weirs, tubes, pipes, and channels is so fully discussed under "Hydraulics", that it only remains here to mention one or two simple methods applicable to small quantities, and to make a few remarks regarding the subject of leakage. In any particular case, that method of determining q is to be selected which will furnish the required degree of precision with the least expense.

For a small discharge, the water may be allowed to fall into a tank of known capacity. The tank should be of uniform horizontal cross-section, whose area can be accurately determined; and then the heights alone need be observed in order to find the volume. These, in precise work, will be read by hook gages; and in cases of less accuracy, by measurement with a graduated rod. At the beginning of the experiment, a sufficient quantity of water must be in the tank so that a reading of the gage can be taken; the water is then allowed to flow in, the time between the beginning and end of the experiment being determined by a stop watch, duly tested and rated. This time must not be short, in order that the slight errors in reading the watch may not affect the result. The gage is read at the close of the test after the surface of the water becomes quiet; and the difference of the gage readings gives the depth which has flowed in during the observed time. The depth, multiplied by the area of the cross-section, gives the volume; and this, divided by the number of seconds during which the flow occurred, furnishes the discharge per second q.

If the discharge is very small, it may be advisable to weigh the water rather than to measure the depths and cross-sections. The total weight divided by the time of flow then gives directly the weight W. This has the advantage of requiring no temperature

observation, and is probably the most accurate of all methods; but unfortunately it is not possible to weigh a considerable volume of water, except at great expense.

When water is furnished to a motor through a small pipe, a common water meter may often be advantageously used to determine the discharge. No water meter, however, can be regarded as accurate until it has been tested by comparing the discharge as recorded by it with the actual discharge as determined by measure-

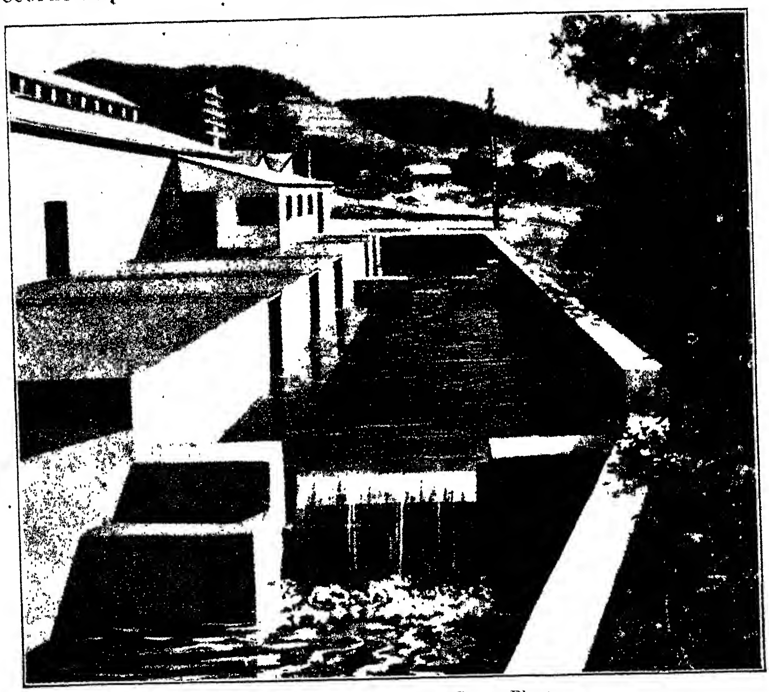


Fig. 75. Weir for Modern Power Plant

ment or weighing in a tank. Such a test furnishes the constants for correcting the results found by its readings, which otherwise are liable to be 5 or 10 per cent in error.

Leakage. The leakage in the flume or penstock before the water reaches the wheel, should not be included in the value of W, which is used in computing its efficiency, although it is needed in order to ascertain the efficiency of the entire plant. The manner of determining the amount of leakage will vary with the particular circumstances of the case in hand. If it be very small, it may

be caught in pails and directly weighed. If large in quantity, the gates which admit water to the wheel may be closed, and the leakage being then led into the tailrace, it may be there measured by a weir, or by allowing it to collect in a tank. The leakage from a vertical penstock whose cross-section is known, may be ascertained by filling it with water, the wheel being still, and then observing the fall of the water level at regular intervals of time. In designing constructions to bring water to a motor, it is best, of course, so to arrange them that all leakage will be avoided; but this cannot often be fully attained, except at great expense.

The most common method of measuring q is by means of a weir placed in the tailrace below the wheel, as in Fig. 75. This has the disadvantage that it sometimes lessens the fall which would be otherwise available, and that often the velocity of approach is high. It has, however, the advantage of cheapness in construction and operation, and for any considerable discharge appears to be almost the only method which is both economical and precise. If the weir is placed above the wheel, the leakage of the penstock must be carefully ascertained.

80. Effective Head on Motor. The total available head H between the surface of the water in the reservoir or headrace and that in the lower pool or tailrace, is determined by running a line of levels from one to the other. Permanent bench marks being established, gages can then be set in the headrace and tailrace, and graduated so that their zero points will be at some datum below the tailrace level. During the test of a wheel, each gage is read by an observer at stated intervals; and the difference of the readings gives the head H. In some cases it is possible to have a floating gage on the lower level, the graduated rod of which is placed alongside a glass tube that communicates with the upper level; the head H is then directly read by noting the point of the graduation which coincides with the water surface in the tube. This device requires but one observer, while the former requires two; but it is usually not the cheapest arrangement, unless a large number of observations are to be taken.

From this total head H, are to be subtracted the losses of head in entering the forebay and penstock, and the loss of head

in friction in the penstock itself, and these losses may be ascertained by the methods discussed in "Hydraulics". Then

$$h = H - h' - h''$$

is the effective head acting upon and chargeable to the wheel. In properly designed approaches, the lost heads h' and h'' are very small.

When water enters upon a wheel through an orifice which is controlled by a gate, losses of head will result, which can be estimated by the appropriate hydraulic formulas. If this orifice is in the headrace, the loss of head should be subtracted, together with the other losses, from the total head H. But if the regulating gates are a part of the wheel itself, as is the case in a turbine, the loss of head should not be subtracted, because it is properly chargeable to the construction of the wheel, and not to the arrangements which furnish the supply of water. In any event that head should be determined which is to be used in the subsequent discussions: if the efficiency of the fall is desired, the total available head is required; if the efficiency of the motor, that effective head is to be found which acts directly upon it.

When water is delivered through a nozzle or pipe to an impulse wheel, the head h is not the total fall, since a large part of this may be lost in friction in the pipe, but is merely the velocity head $\frac{v^2}{2g}$ of the issuing jet. The value of v is known when the discharge q and the area of the cross-section a of the stream have been determined; and

$$h = \frac{v^2}{2g} = \frac{q^2}{2ga^2}$$

It is here assumed that the center of the nozzle is substantially at tailwater level. In the same manner, when a stream flows in a channel against the vanes of an undershot wheel, the effective head is the velocity head; and the theoretical energy is, in either case

$$K = Wh = W\frac{v^2}{2g} = \frac{wq^3}{2ga^2}$$

If, however, the nozzle be above the elevation of tailwater, and the water, in passing through the wheel, falls a distance h_0' below

the mouth of the nozzle, then the head which actually acts upon the wheel considered as a water motor merely, is given by

$$h_1 = \frac{v^2}{2g} + h_0'$$

but the effective head chargeable to the wheel as part of the installation is

$$h = \frac{v^2}{2g} + h_0$$

in which h_0 is the distance of the nozzle center above the tailwater level. In order to utilize the fall h_0 efficiently, it is plain that the wheel should be placed as near the level of the tailrace as possible.

Lastly, when water enters a turbine wheel through a pipe, a piezometer or a pressure gage may be placed near the wheel entrance, to register the pressure head during the flow; if this pressure head, measured from the water level in the tailrace, be called h_0'' and if the velocity in the pipe be v, then

$$h = \frac{v^2}{2g} + h_0''$$

is the effective head chargeable to the wheel as part of the installation. The head chargeable to the wheel itself, regarded merely as a water motor, without reference to its installation, is

$$h_1 = \frac{v^2}{2g} + h_0'$$

in which h_0' is the pressure head measured upward from the lowest part of the exit orifices.

From the above discussion, it will be seen that a distinction is sometimes made between the efficiency of the motor itself, and the efficiency of the motor installed as a hydraulic machine. In the former case, only that part of the available head which is actually utilized by the motor should be used in the calculations; in the latter case, the entire available head. The case of discharge into a draft tube is considered in a subsequent article.

81. Measurement of Effective Power. The effective work and horsepower delivered by a water wheel or hydraulic motor are often required to be measured. Water power may be sold by means of the weight W, or quantity q, furnished under a certain head, leaving the consumer to provide his own motor; or it may

be sold directly by the number of horsepower. In either case, tests must be made from time to time, in order to insure that the quantity contracted for is actually delivered, and is not exceeded. It is also frequently required to measure the effective work, in order to ascertain the power and efficiency of the motor, either because the party who buys it has contracted for a certain power and efficiency, or because it is desirable to know exactly what the motor is doing, in order to improve if possible its performance.

The test of a hydraulic motor has for its object: (1) the determination of the effective energy and power; (2) the determination of its efficiency; and (3) the determination of that speed which gives the greatest power and efficiency. If the wheel is still, there is no power; if it is revolving very fast, the water is flowing through it so as to change but little of its energy into work; and in all cases there is found a certain speed which gives the maximum power and efficiency. To execute these tests, it is not at all necessary to know how the motor is constructed, or the principle of its action. although such knowledge is very valuable, and is in fact indispensable, in order to enable the engineer to suggest methods by which its operation may be improved.

A method by which the effective work of a small motor may be measured, is to compel it to exert all its power in lifting a weight. For this purpose, the weight may be attached to a cord which is fastened to the horizontal axis of the motor, around which it winds as the shaft revolves. The wheel then expends all its power in lifting this weight W_1 through the height h_1 in t_1 seconds; and the work performed per second, then, is

$$k=\frac{W_1h_1}{t_1}$$

This method is rarely used in practice, on account of the difficulty of measuring t_1 with precision.

Dynamometer. Usually, the effective work of a hydraulic motor is gaged by means of the friction brake or power dynamometer invented by Prony about 1780. In Fig. 76 is illustrated a simple method of applying the apparatus to a vertical shaft, the upper diagram being a plan, and the lower an elevation. Upon the vertical shaft is a fixed pulley; and placed against this, are seen two rectangular pieces of wood hollowed so as to fit it, and connected

by two bolts. By turning the nuts on these bolts while the pulley is revolving, the friction can be increased at pleasure, even to the extent of stopping the motion; around these bolts, between the blocks, are two spiral springs (not shown in the diagram) which press the blocks outward when the nuts are loosened. To one of these blocks is attached a cord, which runs horizontally to a small movable pulley over which it passes, and supports a scale pan in which weights are placed. This cord runs in a direction opposite to the motion of the shaft, so that when the brake is tightened it is prevented from revolving by the tension caused by the weights. The direction of the cord in the horizontal plane must be such that the perpendicular let fall upon it from the center of the shaft, or its lever arm, is constant; this can be effected by keeping the small pointer on the brake at a fixed mark established for that purpose.

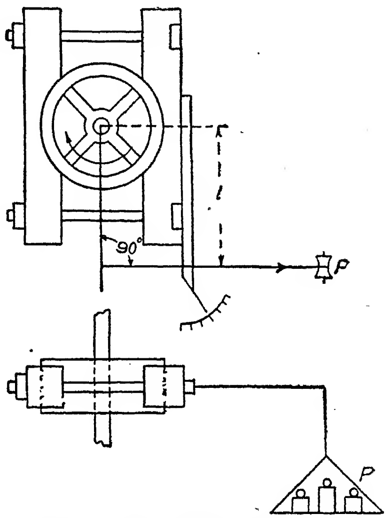


Fig. 76. Measuring Work of Motor by Prony Brake

To measure the work done by the wheel, the shaft is disconnected from the machinery which it usually runs, and is allowed to revolve, transforming all its work into heat by the friction between the revolving pulley and the brake, which is kept stationary by tightening the nuts and at the same time placing sufficient weight in the scale pan to hold the pointer at the fixed mark. Let n be the number of revolutions per second, as determined by a counter attached to the shaft; P, the tension in the cord, which is equal to the weight of the scale pan and its loads; l, the lever arm of this tension with respect to the center of the shaft; r, the radius of the pulley; and F, the total force of friction between the pulley and the brake. Now, in one revolution, the force F is overcome through the distance $2\pi r$; and in n revolutions through the distance $2\pi rn$. Hence the effective work done by the wheel in one second is

$$K = F \times 2\pi rn = 2\pi nFr$$

The force F, acting with the lever arm r, is exactly balanced by the force P acting with the lever arm l; accordingly the moments Fr and Pl are equal; and hence the work done by the wheel in one second is

$$K = 2\pi nPl \tag{101}$$

If P be in pounds, and l in feet, the effective horsepower hp. of the wheel is given by

$$\text{hp.} = \frac{2\pi nPl}{550} \tag{101a}$$

As the number of revolutions in one second cannot be accurately read, it is usual to record the counter readings every minute or half minute. If N be the number of revolutions per minute,

$$\text{hp.} = \frac{2\pi NPl}{33{,}000} \tag{101b}$$

It is seen that this method is independent of the radius of the pulley, which may be of any convenient size. For a small motor, the brake may be clamped directly upon the shaft; but for a large one a pulley of considerable size is needed, and a special arrangement of levers is used, instead of a cord.

The efficiency of the motor is now found by dividing the effective work per second by the theoretic work per second. Let K be this theoretic work, which is expressed by Wh; then

$$e = \frac{k}{K} \qquad \text{or, } e = \frac{\text{hp.}}{\text{HP.}} \tag{101c}$$

The work measured by the friction brake is that delivered at the circumference of the pulley, and does not include that power which is required to overcome the friction of the shaft upon its bearings. The shaft or axis of every water wheel must have at least two bearings, the friction of which consumes probably about 2 or 3 per cent of the power. The hydraulic efficiency of the wheel, regarded as a user of water, is hence 2 or 3 per cent greater than the computed volume of e.

There are in use various forms and varieties of the friction brake, but they all act upon the principle and in the manner above described. For large wheels, they are made of iron, and almost completely encircle the pulley; while a special arrangement of levers is used to lift the large weight P. If the work transformed

into friction be large, both the brake and the pulley may become hot, to prevent which a stream of cool water is allowed to flow upon them. To insure steadiness of motion it is well that the surface of the pulley should be lubricated, which, for a wooden brake, is well done by the use of soap. It is important that the connection of the cord to the brake should be so made that the lever arm l increases when the brake moves slightly with the wheel; if this is not done, the wheel will be apt to cause the brake to revolve with it.

TESTING*

82. Holyoke Testing Flume. At Holyoke, Massachusetts, where the Connecticut River furnishes a large water power, falling some 60 feet, the Holyoke Water Power Company controls the water rights, and leases power to the many mill operators of that city. The mill owners pay a certain price per annum per *mill-power,* which, in that locality, is the right to use 38 cubic feet of water per second under a head of 20 feet, either for continuous use (a 24-hour day) or for a definite fraction of each day.

In order that the rate at which any mill turbine uses water at any stage or position of its gate or regulating apparatus may become known by simply observing the position of the gate, each turbine, before being installed in the mill where it is to work, is tested at the testing flume of the company, and thus becomes a water meter, whose indications, when the motor is in final place, are noted from day to day by an inspector of the company. In the same test, its power, best speed, and efficiency are also determined.

Features. The flume occupies the lower part of a substantial building; the main features are shown in vertical section in Fig. 77. The walls of the wheel pit *DD*, which is 20 feet square, are built of stone masonry, and lined with brick laid in cement. The water is admitted to it from the head canal through a trunk or penstock, and vestibule, which are not shown in the figure. Over an opening in the floor of the wheel pit, the wheel *W* to be tested is set in place, the water discharged from it finding its way through a large opening into the tailrace *C*, 35 feet long and 20 feet wide; and finally over

* Articles 82 to 85 inclusive have been taken, with slight changes, from Professor Irving P. Church's "Hydraulic Motors", John Wiley & Sons, New York.

a sharp-crested weir at A, into the lower canal. The whole head h available for testing may be from 4 to 18 feet for the smaller wheels, and from 11 to 14 feet for large wheels, up to 300 horsepower. The measuring capacity of the weir, which may be used to its full length, 20 feet (and then would have no end contractions), is about 230 cubic feet per second. The head h becomes known in any test by observations of the water level in two glass tubes communicating with the respective bodies of water W and C. The water in channel C, which is a channel of approach for the weir A, communicates (at a point some distance back of the weir) by a lateral pipe with the interior of a vessel open to the air, in a side chamber. Water rises in this vessel, and finally remains stationary at the same level as that of the surface in the channel of approach.

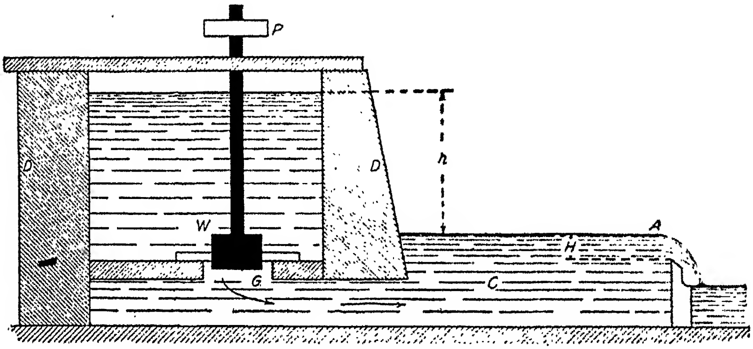


Fig. 77. Vertical Section of Holyoke Testing-Flume

A hook gage being used in connection with this vessel, observations and readings are taken, from which the value of H, or *head on the weir*, may be computed, for use in the proper weir formula for the discharge q.

Fig. 77 shows a turbine in position for testing, with a vertical shaft—the ordinary case. Upon the upper end of the shaft is secured a cast-iron pulley P, to the rim of which the Prony brake is fitted for purposes of test.

Procedure. The procedure of testing follows: The brake being carefully balanced and adjusted beforehand, a light weight was placed on the scale pan, and the wheel started at full gate; sufficient friction was then produced to balance the weight, and the speed of wheel noted. The load was then increased at intervals of 2 or 3

TREMONT TURBINE TEST

(Selected Experiments)

0	1	2	3	4	5	6	7
Gate Opening	No. of Exper.	h (Feet.)	n (Rev. per sec.)	q (Cu. ft. per sec.)	$2\pi n\ Pl$ (Ft.-lb. per sec.)	e (Effy.)	hp.
Full	1	12.80	0.00	135.6	0	0.00	
Full	2	12.95	0.45	133.4	73,160	.68	
Full	3	12.97	0.53	133.7	78,490	.72	
Full	4	12.97	0.60	134.8	82,110	.75	
Full	5	12.94	0.64	135.1	83,960	.77	
Full	**6**	**12.90**	**0.85**	**138.2**	**88,210**	**.794**	**160.3**
Full	7	12.90	0.88	139.0	88,190	.788	
Full	8	12.90	0.90	139.6	88,076	.784	
Full	9	12.85	1.00	141.9	86,310	.75	
Full	10	12.85	1.06	142.5	83,970	.73	
Full	11	12.80	1.18	144.8	77,150	.67	
Full	12	12.70	1.31	147.3	66,840	.57	
Full	13	12.65	1.46	152.3	51,680	.43	
Full	14	12.55	1.60	156.6	33,350	.27	
Full	15	12.54	1.79	162.3	0	0.00	
Part	16	13.51	0.00	60.3	0	0.00	
Part	17	13.55	0.46	67.8	24,460	.43	
Part	**18**	**13.48**	**0.67**	**71.8**	**27,980**	**.46**	**50.9**
Part	19	13.39	0.96	76.6	21,250	.33	
Part	20	13.34	1.25	80.4	0	.00	

minutes, by 25 pounds at a time, until the speed of the wheel had fallen below that of maximum efficiency for the head; the weights were then reduced again, and the velocity of the wheel allowed to increase until the maximum was again passed. The same process was then repeated within a smaller range of speed and with smaller variations of load, until the speed of best work had been more exactly ascertained, and the performance of the turbine at maximum efficiency, under full head and at full gate, had been very precisely determined. This was repeated at each of the part gates, usually down to one-half maximum discharge.

83. Test of Tremont Turbine. The test of the Tremont turbine, a 160-horsepower turbine of the radial outward-flow type (Fourneyron), made at Lowell, Massachusetts, in 1855, by Mr. J. B. Francis, was an event of special interest in the history of hydraulic science, and has become classic. Though the test is by no means recent, it was carried out so thoroughly as to make its details highly

instructive to the student of hydraulics. The main features of this test will now be presented and commented on.

Data. Inner and outer radii of turbine were 3.37 feet and 4.14 feet, respectively; height between crowns, 0.937 foot at entrance, and 0.931 foot at exit. There were 33 guide blades and 44 turbine vanes. As to angles, $\alpha=28$ degrees, $\phi=90$ degrees, and $\beta=22$ degrees; and the head h on the wheel varied from about 12.5 feet to about 13.5 feet. The gate was a thin cylinder, movable vertically between the guides and the wheel. There were no horizontal partitions dividing the wheel channels—in fact, no special device for preventing the loss of head usually arising at part gate with this kind of regulating apparatus.

The annexed tabular matter gives the principal data and results of Mr. Francis's test of the Tremont turbine, arranged in the order of the speed of wheel. In Experiments Nos. 1 to 15 (see column 1), the cylindrical gate was fully open (full gate); while in Experiments 16 to 20, it was in a single fixed position, leaving open, at the wheel entrance, about one-quarter of the vertical height between crowns; in other words, the gate was drawn up about one-quarter of its full range of height. In this special part-gate position, however, the quantity of water passing per second was much greater than one-quarter of that passing at full gate, as is seen from the values of q in column 4. For example, in Experiment 18, in which (for this position of the gate) the efficiency was a maximum, the value of q is about one-half of the q used in Experiment 6, which gives the maximum efficiency at full gate. It would be said, therefore, that in Experiment 18 the wheel was working at about half gate. The heading of each column of the table shows clearly the nature of the quantity given in that column, and the units of measurement involved in its numerical value.

The rate of flow, or discharge in cubic feet per second, was measured by two weirs at the end of the tailrace, using the Francis weir formula; and the useful power was measured by means of the Prony brake, which in this case consisted of a large and strong friction brake with arcs of wood rubbing on the cast-iron pulley which was keyed to the turbine shaft, and arranged with a bell-crank lever and dashpot to cushion the motion of the lever. In this brake, $r=2.75$ feet, and $l=10.83$ feet.

Typical Computations. To explain the computations, Experiment No. 6 will be selected as typical. In this test, 1524 pounds were placed in the scale pan, and the nuts tightened up, until the wheel raised this weight P and held it just balanced. When the speed of the wheel had adjusted itself to the load, the speed counter indicated $n=0.851$ revolution per second. Hence

$$
\begin{aligned}
K &= 2\pi n P l \\
&= 2\times 3.1416\times 0.851\times 1524\times 10.83 \\
&= 88{,}214 \text{ ft.-lb. per sec.} = 160.3 \text{ hp.}
\end{aligned}
$$

For computing the value of the discharge q, it is to be observed that the water passed over two contracted weirs with a combined length $b=16.98$ feet; the number of end contractions was therefore $n=4$; the head over the weir crest was $H=1.87$ feet; velocity of approach was not considered. Therefore

$$
\begin{aligned}
q &= 3.33\ (b-0.1nH)\,H^{\frac{3}{2}} \\
&= 3.33\ (16.98-.75)\ (1.87)^{\frac{3}{2}} \\
&= 138.2 \text{ cu. ft. per sec.}
\end{aligned}
$$

The difference in elevation between headwater and tailwater levels in this experiment was 12.90 feet; consequently the total available energy was

$$K=138.2\times 62.5\times 12.90=111{,}400 \text{ ft.-lb. per sec.}$$

Therefore the efficiency was

$$e=\frac{88{,}214}{111{,}400}\doteq 79.4 \text{ per cent}$$

84. Discussion. *Results of Preceding Test.* In the experiments with full gate, Nos. 1 to 14 inclusive (see tabulation), on account of the progressive lessening of the weight P in the scale plan (the brake friction being regulated each time to correspond), the uniform speed to which the wheel adjusts itself in successive experiments increases progressively from the zero value, or state of rest, of Experiment 1 (when the friction was so great as to prevent any motion), up to a maximum rate of 1.79 revolutions per second, attained when no brake friction whatever (no load) was present. In this last experiment, there being no useful work done, all the energy of the mill site is wasted, partly in axle friction, but chiefly in fluid friction (eddying of the water, and finally, heat), both in the wheel passages and also in the tailrace, where the water which

has left the wheel with high velocity soon has its velocity extinguished. The same statement is true also for Experiment No. 1, except that axle friction is wanting. In both experiments the efficiency is, of course, zero.

The quantity of water discharged per second q is seen to increase slowly (after Experiment 2) from 133.4 to 162.3 cubic feet per second, though not differing from the average by more than 10 per cent. This may be accounted for, in a rude way, as an effect of centrifugal action (as in a centrifugal pump), since the Tremont turbine is an outward-flow wheel. The reverse is found to be true for inward-flow turbines, notably the Thompson vortex wheel, which is therefore to some extent self-regulating in the matter of speed, since a less discharge at a speed higher than the normal diminishes the power, and hence the tendency to further increase of speed.

In the succession of experiments Nos. 1 to 15 (all at full gate and under practically the same head h), the efficiency is seen to have a zero value both at beginning and end of this series, and to reach its maximum at about the sixth experiment, in which the speed is noted as being about one-half that at which the turbine runs when entirely unloaded (Experiment 15). This is roughly true in nearly all turbine tests; but a notable feature of considerable practical advantage is that a fairly wide deviation from the best speed affects the efficiency but slightly. For instance, a variation of speed by 25 per cent either way from the best value (0.85 revolution per second) causes a diminution in the efficiency of only about 4 per cent.

It should be remembered, also, in this connection, that since the water used per second q is somewhat different at different speeds (at full gate), the speed of maximum power differs slightly from that of maximum efficiency.

In the five part-gate experiments, Nos. 16 to 20, the gate remains fixed in a definite position (about one-quarter raised, although the discharge is about one-half that of full gate) through all these five runs. The head is practically constant. At first the wheel is prevented from turning. The power and efficiency are then, of course, zero; but $q=60.3$ cubic feet per second. As the turbine is permitted to revolve under progressively diminishing friction, the

speed of steady motion becomes greater, reaching its maximum (1.25 revolutions per second) when the wheel runs unloaded, in Experiment 20; but the power reaches a maximum and then diminishes. The same is true of the efficiency, whose maximum (in Experiment 18) is seen to be about 46 per cent only. This forms a striking instance of the disadvantage and wastefulness of a cylindrical gate unaccompanied by other mitigating features, when in use at part gate. This defect, however, may be largely remedied by the use of horizontal partitions in the wheel channels, as in Fig. 89, page 136, or by employing curved upper crowns, as in the "American" inward-

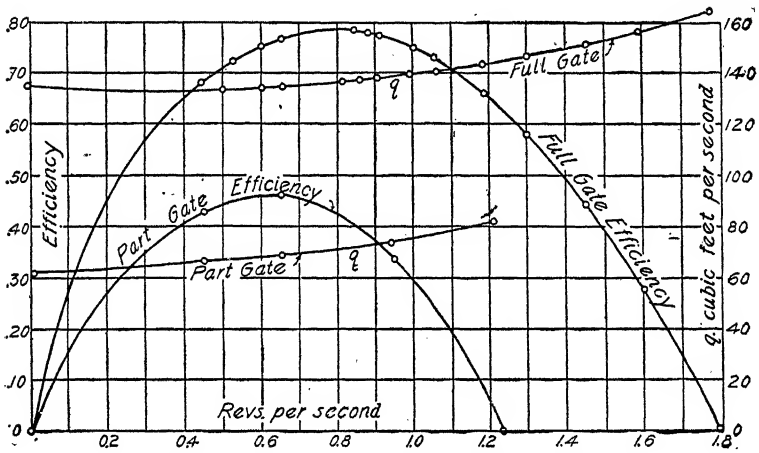


Fig. 78. Curves Showing Results of Test of Tremont Turbine

and-downward turbines. Fig. 78 is a graphic representation of the results of the Tremont turbine tests just discussed.

Leffel Turbine Test. At the station of the Telluride Power Transmission Company a Leffel turbine was tested in place by Chief Engineer P. N. Dunn, under a working head varying between 210 feet and 212 feet, approximately, the results of which are shown graphically in Fig. 79.

85. Value of Testing.* It is now well known that the early development of the hydraulic turbine in America was carried along empirical lines, and that the application of logical engineering methods has been of comparatively recent date. It is also known

* Abstracted from an article by H. B. Taylor, hydraulic engineer, in *General Electric Review*, June, 1914.

that many turbine manufacturers in this country still adhere to the older methods, and produce turbines of stock designs and standard sizes, relying to a great extent on the use of existing patterns.

The "stock turbine", so-called, has not for many years been representative of the best work produced in the United States. A number of American builders have successfully undertaken development requiring the highest type of engineering skill and initiative, and much pioneer work in the extension of hydraulic turbines, to

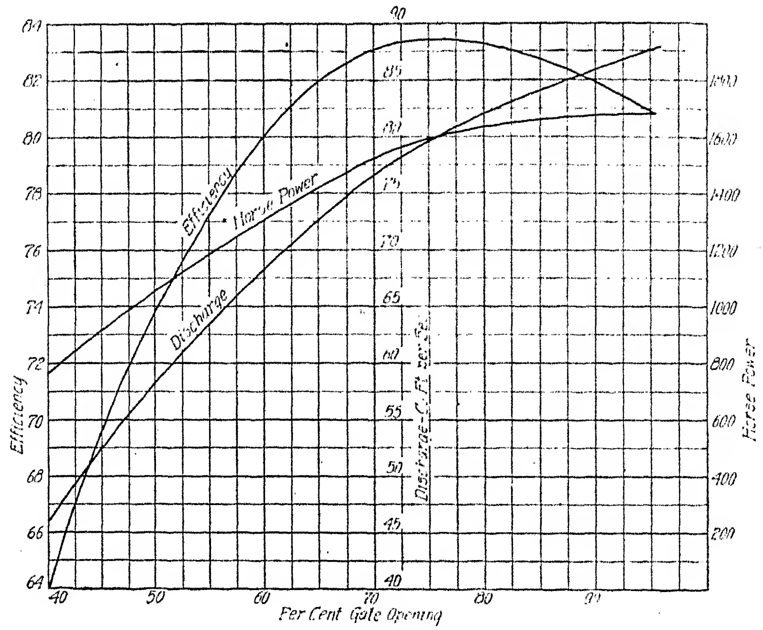


Fig. 79. Curve Showing Results of Tests on Leffel Turbine

cover most severe conditions and a wide range of capacity and dimensions, has been undertaken.

"A large part of the credit for present attainments in American turbine construction has been attributed to the systematic testing of water wheels, which has always been a feature of American practice. Several thousand tests have been made at the Holyoke testing flume, and undoubtedly the results of these experiments have been of the greatest service to American engineers. Until these experimental results were supplemented by proper theoretical investigation, progress was, however, comparatively slow, and recent progress

has been possible only through a more scientific handling of the problems which have arisen."

Results of Model Tests. Discussing the question whether the results of tests on small models are of value in predicting the performance of the full-sized turbines, in place, the author reaches the following conclusions:

"(1) If results of the Holyoke tests on a small model runner are properly interpreted and applied scientifically to the design of a large turbine of like characteristics, then the results with respect to efficiency and power obtained from the large turbine when tested in

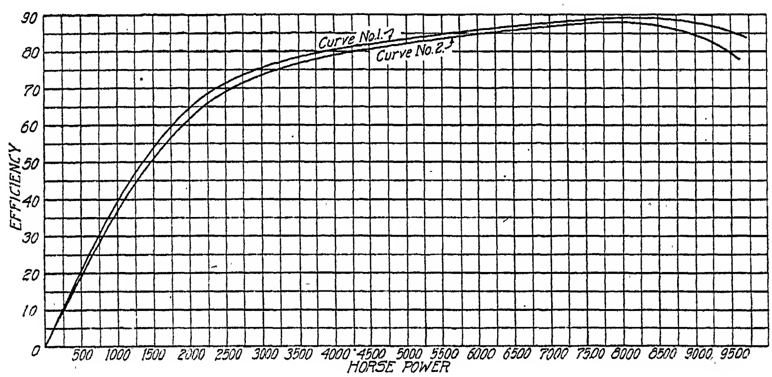


Fig. 80. Curves Showing Comparison of Official Efficiency Test for Washington Water Power Company, 9000 HP. Turbines (Curve No. 1) with Test of Experimental Runner at Holyoke Flume (Curve No. 2)
Courtesy of I. P. Morris Company, Philadelphia, Pennsylvania

place will be found to check, within a very small percentage, the results forecast by the Holyoke test.

"(2) The variation of efficiency obtained in place from that shown at Holyoke may be calculated with a satisfactory degree of precision. If the setting of the large turbine is equivalent to that of the model when tested at the Holyoke flume, the shape of the efficiency-horsepower curve of the large turbine will check the corresponding Holyoke curve very closely—see Figs. 80 and 81.

"(3) If the setting of the large unit is superior to that in which the model runner was tested at Holyoke, as is almost always the case in larger modern developments, then the efficiencies of the large machine will exceed those obtained at Holyoke, and the effect of increased dimensions of the runner of the large turbine over the

model runner, as well as the variation in head, can be definitely computed without any notable degree of uncertainty.

"(4) The effect of increases in head and dimensions on turbines of exactly similar design is not to impair the efficiency, as might be inferred from the above quotations, but both changes have the same result, viz, an increase in efficiency if the character of the turbine setting remains the same.

"It may be stated that in every instance where a large turbine has failed to produce results as good as those forecast by the Holyoke tests, such failure may be attributed either to improper appli-

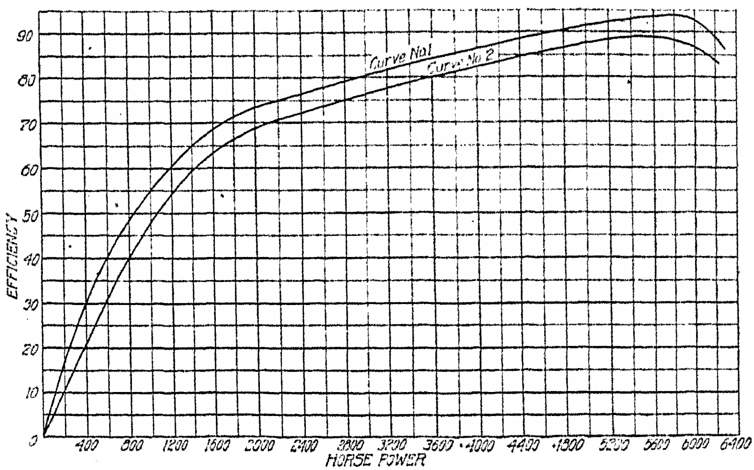


Fig. 81. Curves Showing Comparison of Official Efficiency Test Appalachian Power Company, 6000-HP. Turbines (Curve No. 1) with Test of Experimental Runner at the Holyoke Flume (Curve No. 2)

cation of the tests or to the placing of too much reliance on the runner without proper regard to the effect upon the runner of defective wheel casings and draft tubes. If engineers permit the runners of large turbines to be used in connection with defective settings, then the failure of the large turbines to come up to expectations should not reflect on the Holyoke flume.

"It is not impossible, as may be inferred, to predict from tests at a low head on a small model runner of homologous design, the performance of a large turbine under a high head."

The preceding statements are in line with accepted principles and in accord with both theory and experience. For example, in

Fig. 80 are shown two comparative efficiency curves, one of a regularly installed commercial turbine, and the other of the small-sized experimental turbine of the same design.

The upper curve, labeled curve No. 1, represents the efficiency test curve of the 9000-horsepower turbines* built for the Washington Water Power Company. These turbines operate under 66-foot head at a speed of 150 revolutions per minute with rated runner diameter of 6 feet 2 inches, and are of the horizontal-shaft two-runner central-discharge type with volute casings. Curve No. 2 indicates the efficiency obtained at the Holyoke test, in vertical setting, of the experimental runner having a rated diameter of 2 feet $8\frac{13}{16}$ inches.

It will be noted that the curves in Fig. 80 are almost identical in shape; the efficiency of the large units exceeding by a small margin that of the experimental runner. This illustrates the fact that in the case of large units, with properly designed water passages, even when the type is entirely different from that of the experimental model, an accurate forecast of the performance of the large unit may be secured from the test of the model. Although there were losses in the draft chest of the large turbines which were not present in the experimental testing outfit, these losses were not sufficient to overcome the gain in efficiency in the other water passages.

Another instance is illustrated in Fig. 81, where exceptionally high efficiency has been realized in the full-sized units. Here, Curve No. 1 represents the official test curve of the 6000-horsepower turbines* built for the Appalachian Power Company, where the turbines are of the single-runner vertical-shaft type operating under 49-foot head at a speed of 116 revolutions per minute and with rated runner diameter of 7 feet $6\frac{1}{4}$ inches. Curve No. 2 in this connection is derived from the test at Holyoke of the small experimental runner with rated diameter of $27\frac{5}{8}$ inches. Although not coincident, both curves are practically identical in shape, but owing to better arrangement of water passages in the Appalachian plant (see section in Fig. 123), the efficiencies of the large units exceed those of the experimental runner. It will be noted that at the maximum point in the curve the margin of difference is 5 per cent.

* Designed and constructed by the I. P. Morris Company, Philadelphia, Pennsylvania.

REACTION AND IMPULSE TYPES*

ANALYSIS OF REACTION TURBINES

86. **Inward Flow or Outward Flow.** *Discharge.* The following analysis applies equally well with either direction of flow. The discharge from a reaction turbine, unlike that from an impulse turbine, depends on the speed of revolution, as well as on the orifice areas, as in the case of the reaction wheel already discussed. Let Fig. 82 represent diagrammatically an outward-flow turbine with the customary notation as shown in Fig. 83 and explained in preceding analyses. In addition, let a_1 a, and a_0 be the areas of the respective orifices or water passages, measured normal to the directions of V_1, V, and v_0; and let H represent the pressure head on the guide orifices at the gate openings, as would be indicated

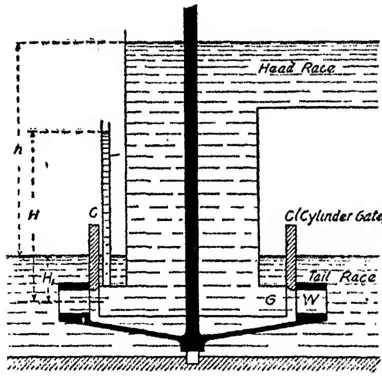


Fig. 82. Diagrammatic Representation of Outward-Flow Turbine

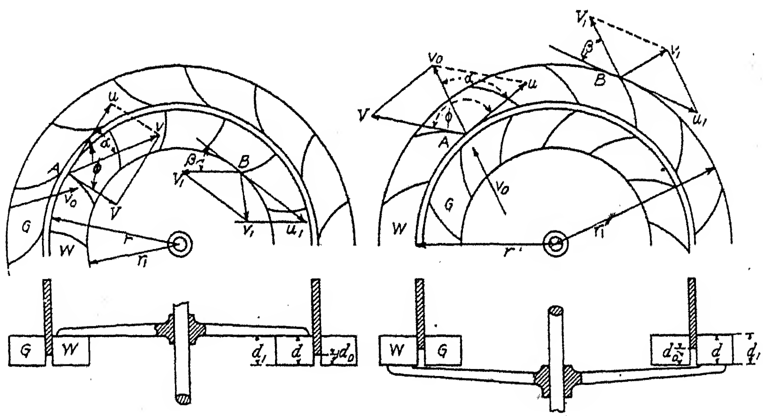


Fig. 83. Diagram Showing Customary Turbine Notations for Inward and Outward Flow

by piezometer tubes or pressure gages if they were inserted at such points.

*The analytical treatment throughout this work follows closely that of Professor Mansfield Merriman, "Treatise on Hydraulics", John Wiley and Sons, New York.

From Article 5, Part I, neglecting frictional losses,

$$H+\frac{v_0^2}{2g}=h+H_1$$

also, from Article 69, Part I, neglecting frictional losses,

$$H_1+\frac{V_1^2}{2g}-\frac{u_1^2}{2g}=H+\frac{V^2}{2g}-\frac{u^2}{2g}$$

The addition of these two equations results in the following formula

$$V_1^2-V^2+v_0^2=2gh+u_1^2-u^2 \qquad \textbf{(102)}$$

Being a reaction turbine, the buckets are completely filled; therefore the same quantity of water must pass per second through each of the areas a_1, a, and a_0; from which condition the following relations are obtained.

$$V_1=\frac{q}{a_1}; \quad V=\frac{q}{a}; \quad v_0=\frac{q}{a_0}$$

Substituting these values in the last formula above, and solving for q, there results

$$q=c_d\sqrt{\frac{2gh+u_1^2-u^2}{\frac{1}{a_1^2}-\frac{1}{a^2}+\frac{1}{a_0^2}}}=m\sqrt{2gh+u_1^2-u^2} \qquad \textbf{(102a)}$$

which is a formula for discharge through a reaction turbine, for either inward or outward flow. The coefficient c_d is introduced to take account of losses through leakage and friction, which factors were not considered in the above analysis. For an outward-flow turbine, u_1 is greater than u, consequently the discharge increases with the speed; for an inward-flow turbine u_1 is less than u, and therefore the discharge varies inversely with the speed.

The value of the coefficient c_d varies with the head of water and with the details of the wheel design. In one case of an outward-flow turbine—in which $r=2.67$ feet; $r_1=3.32$ feet; with total head of water varying between 17.16 and 17.34 feet; number of revolutions per minute, between 63.5 and 100; and the discharge with full gate, from 117 to 127.7 cubic feet per second—the value of c_d was found to range between .941 and .950.

A formula for discharge from a turbine operating at part gate is difficult to formulate theoretically, because of the losses of head resulting from the partial closure, analytical expressions for which

are not definitely known for turbines. The values of q for any turbine operating at part or full gate may be obtained by measuring the quantity of water actually discharged, by any of the various methods described in works on Hydraulics, and substituting the resulting values of q in Equation (**102a**).

Work and Efficiency. The following analysis is also valid for both directions of flow. In addition to the notation adopted and employed in the preceding article, and indicated on the corresponding diagrams, let d_1, d, and d_0 be the respective depths of the exit, the entrance, and the guide orifices or water passages. With gates fully open (in which case d_0 becomes equal to d), and neglecting the thickness of the vanes or passage walls,

$$a_0=2\pi rd\sin\alpha;\ a=2\pi rd\sin\phi;\quad a_1=2\pi r_1d_1\sin\beta$$

Since, in a reaction turbine, the passageways are always completely full of water,

$$q=v_0\times 2\pi rd\sin\alpha=V\times 2\pi rd\sin\phi=V_1\times 2\pi r_1d_1\sin\beta \qquad \textbf{(102b)}$$

The general conditions previously established and discussed—that for maximum efficiency the water must enter tangentially to the vanes, and the absolute velocity of the water at discharge must be as low as possible—will be fulfilled for the first case, when u and v_0 are proportional to the sines of their opposite angles—that is, when

$$\frac{u}{v_0}=\frac{\sin(\phi-\alpha)}{\sin\phi} \qquad \textbf{(103)}$$

and for the second case, approximately and simply, when

$$u_1=V_1$$

(Theoretically, $V_1=u_1\cos\beta$, as explained in a preceding section.)

Substituting u_1 for V_1 in the third member of Equation **102b** and equating it to the first, there results

$$\frac{u_1}{v_0}=\frac{rd\sin\alpha}{r_1d_1\sin\beta}$$

Then, since $\frac{u}{u_1}=\frac{r}{r_1}$, the above equation may be reduced to

$$\frac{u}{v_0}=\frac{r^2d\sin\alpha}{r_1^2d_1\sin\beta} \qquad \textbf{(103a)}$$

From the trigonometric relations at the point A, in Fig. 83,

$$V^2=u^2+v_0^2-2uv_0\cos\alpha$$

Substituting this value of V^2 in Equation (102), and making $V_1 = u_1$, as established above for one of the conditions of maximum efficiency, the following additional important relation immediately results

$$uv_0 = \frac{gh}{\cos\alpha} \qquad \textbf{(104)}$$

From the above necessary relations, the following practical formulas may be developed by combining Equations (103) and (104).

$$u = \sqrt{\frac{gh \sin(\phi - \alpha)}{\cos\alpha \sin\phi}} \qquad \textbf{(105)}$$

and,

$$v_0 = \sqrt{\frac{gh \sin\phi}{\cos\alpha \sin(\phi - \alpha)}} \qquad \textbf{(106)}$$

Equation (105) gives the advantageous velocity of the circumference at the point of wheel entrance, from which the advantageous velocity of the circumference at exit may be obtained from the relation $\frac{u}{u_1} = \frac{r}{r_1}$; and Equation (106) gives the value of the absolute velocity of entrance of the water into the wheel.

By combining Equations (103) and (103a), there is obtained

$$\frac{\sin(\phi - \alpha)}{\sin\phi} = \frac{r^2 d \sin\alpha}{r_1^2 d_1 \sin\beta} \qquad \textbf{(107)}$$

which establishes the necessary relations between the dimensions and angles of the wheel which must obtain in order that the above conclusions may be valid.

The work imparted to the wheel is, theoretically

$$\text{Work} = W\left(h - \frac{v_1^2}{2g}\right) \qquad \textbf{(108)}$$

and the efficiency is theoretically

$$e = 1 - \frac{v_1^2}{2gh} \qquad \textbf{(108a)}$$

in which h indicates the available head properly chargeable to the machine as installed.

By means of Equations (105) and (107), and the relations at the point B, the above value of the efficiency may be reduced to

$$e = 1 - \frac{d}{d_1} \tan\alpha \tan\tfrac{1}{2}\beta \qquad \textbf{(109)}$$

in which d and d_1 are the entrance and exit depths, respectively.

The discharge is

$$q = a_0 v_0$$

and the useful work of the wheel is

$$\text{Work} = e \times wqh$$

It is to be observed that the foregoing formulas for work and efficiency do not take into account losses of energy incurred during the passage of the water to and through the guide and wheel buckets and those due to clearance.

If losses due to impact and friction in the runner buckets be neglected, then the work imparted to the wheel is

$$\text{Work} = W\frac{h' - v_1^2}{2g} \tag{109a}$$

and the efficiency is

$$e = \frac{h' - v_1^2}{2gh} \tag{110}$$

in which h' is the head representing the total energy of the water as it enters the runner buckets.

87. Downward or Parallel Flow. Let r be the mean radius, and u the mean velocity of the entrance and exit orifices of the wheel, and let d and d_1 be the widths of the entrance and exit orifices, respectively. The formulas developed in the preceding articles for inward- and outward-flow reaction turbines may be adapted to this case by making $u_1 = u$, and $r_1 = r$.

Thus the advantageous velocity of entrance is

$$v_0 = \sqrt{\frac{gh \sin \phi}{\cos \alpha \sin (\phi - \alpha)}} \tag{111}$$

The advantageous speed is

$$u = \sqrt{\frac{gh \sin (\phi - \alpha)}{\cos \alpha \sin \phi}} \tag{112}$$

The necessary relation between the vane angles and the wheel dimensions is

$$\frac{\sin (\phi - \alpha)}{\sin \phi} = \frac{d \sin \alpha}{d_1 \sin \beta} \tag{113}$$

and the hydraulic efficiency is

$$e = 1 - \frac{d}{d_1} \tan \alpha \tan \tfrac{1}{2} \beta \tag{114}$$

IMPULSE TURBINE

88. Formulas. The velocity v_0 with which the water leaves the guide orifices and enters the runner buckets, is, in the case of impulse turbines, theoretically equal to $\sqrt{2gh_0}$, where h_0 stands for the effective head on such orifices. The analysis and conclusions of Article 61, Part I, apply directly to such motors. Accordingly, for a properly designed impulse turbine, the entrance angle should be double the approach angle; that is $\phi = 2\alpha$. The advantageous speed is

$$u = \sqrt{\frac{gh_0}{2\cos^2\alpha}}$$

When the motor is running at this best speed, the absolute velocity of exit is

$$v_1 = v_0 \frac{r_1 \sin\frac{1}{2}\beta}{r\cos\alpha}$$

The work imparted to the wheel is

$$\text{Work (Max.)} = W\frac{v_0^2 - v_1^2}{2g} = Wh_0\left[1 - \left(\frac{r_1\sin\frac{1}{2}\beta}{r\cos\alpha}\right)^2\right]$$

and the efficiency is

$$e\ (\text{Max.}) = \frac{h_0}{h}\left[1 - \left(\frac{r_1\sin\frac{1}{2}\beta}{r\cos\alpha}\right)^2\right]$$

in which h is the available head properly chargeable to the motor.

It is clearly to be seen that both the approach angle α and the exit angle β should be small for high efficiency, and that the angle β exercises a greater influence on the efficiency than the angle α, both of which conclusions have already been discussed in the articles referred to above.

The discharge is

$$q = a_0 v_0 = a_0\sqrt{2gh_0}$$

and the work of the turbine per second is

$$\text{Work} = e \times wqh$$

With both reaction and impulse turbines, when the guide buckets are considered part of the turbine, h is the head representing the total available energy existing in the water as it enters the guides. When the turbine runner alone is considered to constitute

the motor proper, and the guides part of the approach, h is the head representing the total available energy existing in the water as it emerges from the guide buckets. In the latter case, in the above analysis, h may be put equal to h_0. In both cases, however, there is a certain amount of loss in the clearance.

89. Definition of Terms. *Admission—Complete and Partial.* These terms signify, respectively, that the feed water is admitted to all or to some of the wheel passages simultaneously.

Gate Opening—Part Gate and Full Gate. These terms refer to the area left open by the regulating mechanism for the passage of the feed water. *Full gate* signifies that the passages are entirely unrestricted, and *part gate* is usually expressed as the proportional part of the full gate area which is free. Occasionally these terms are employed to express the quantity of water flowing at any particular setting of the gate compared with the quantity flowing at full gate; but this use of the terms is misleading and should be avoided.

Clearance. In order to permit freedom of rotation a clear space must be left between the runner and the guide ring; this space is called the *clearance.*

COMPARISON

90. Partial Operation. Reaction turbines operate with complete admission, and part-gate operation entails loss of energy due to contraction and subsequent expansion of stream section. Impulse turbines may have complete or partial admission, the latter condition involving very little sacrifice of energy. Moreover, the regulating devices for impulse turbines are of simpler construction than those of reaction turbines.

To materially reduce the loss of power incidental to the operation of a reaction turbine at part gate, the runner is sometimes constructed with horizontal partitions dividing it into several parts or stories, as described later, Figs. 89 and 90. The gate completely closes off one or more of these divisions during operation at part gate, leaving the remaining divisions only in action. Contraction and subsequent expansion of the stream being thus avoided, the efficiency of partial operation will not be materially reduced.

The "Duplex" motor, a double turbine of the parallel-flow type, consists of a pair of concentric runners in one piece, with a pair of

concentric annular supply guide passages for the feed water. With full operation both sets are in action; for partial operation one set is shut off, the other operating independently without sacrifice of efficiency. Such motors must be distinguished, on the one hand, from double motors of the two-story type, as described in the preceding article, and, on the other hand, from those double turbines in which two essentially separate wheels are combined by mounting on the same shaft for the purpose of increasing the turbine capacity without increasing its diameter, as in the Leffel and other types.

91. **Kinetic Energy.** In an impulse turbine the energy of the jet of water flowing freely (i.e., under atmospheric pressure) with a velocity due to the head is converted into kinetic energy at the inlet surface, and imparts this energy to the curved vanes of the wheel through the dynamic pressure due to its impulse. The passages should never be completely filled, and in order to prevent the formation of eddies at the backs of the vanes, and to insure an unbroken stream through the passages, the sides of the wheels are provided with ventilating holes.

In a reaction turbine only a portion of the available energy is converted into kinetic energy at the inlet surface of the wheel, which is thus impelled by the dynamic pressure of the flowing water, itself under static pressure.

Perhaps the main disadvantages of the impulse type are due to the fact that these turbines must operate in the air with consequent loss of head; or, in case a draft tube is used, an air-admission valve is necessary. In the latter instance, the buckets must be ventilated, and the surface of the water in the draft tube must not be allowed to rise above the level of the exit orifices or above the moving parts of the motor. This is fully explained in a subsequent section.

92. **Speed.** Since an impulse turbine may vary in diameter between wide limits without material hydraulic disadvantage, it follows that in the case of high head and small water supply a comparatively large wheel may be employed at low speed. Under these conditions the speed of a reaction turbine would be too great; and an attempt to reduce the speed by increasing the diameter would materially increase the friction and introduce difficulties in the

proper design of the vanes. With ample water supply and moderate heads the reaction turbine is usually chosen; on the other hand with the opposite conditions a turbine of the impulse type is to be preferred.

The advantageous speed of an impulse turbine remains the same for all positions of the gate; but with reaction turbines it is less at part gate than at full gate; and it has already been stated that the partial closing of the gates results in a material loss of energy in the case of reaction wheels; and since, for many industrial purposes, it is essential to maintain a constant speed in spite of variations in power or useful work, it follows that to maintain this constant speed with a reaction turbine involves considerable loss of efficiency It is also evident that a turbine of the impulse type has a marked advantage in point of efficiency in all cases where the supply of water is low.

93. Entrance and Discharge. In impulse turbines, the entrance angle ϕ should be double the approach angle α; but in reaction turbines it is often greater than 3α, and its value depends on the exit angle β; hence the vanes in impulse turbines are of sharper curvature for the same values of α and β. β is usually greater for inward-flow than for outward-flow reaction wheels, in order that the exit orifices may be sufficiently large. If the entrance angle ϕ is 90 degrees (a good value), Equation (**105**) shows that the velocity u is that due to one-half the head. Equation (**109**) shows that the efficiency is increased by making the exit depth d_1 greater than the entrance depth d (bell-mouthed profiles; diffuser); but usually they do not differ very much, and frequently they are made equal.

Impulse turbines always discharge into the air, at some distance above tail water; consequently that part of the available head between the center of discharge and the tailwater level is lost, unless the motor is provided with a draft tube, in which case only part of the head is lost, as described later. A reaction turbine may discharge into the free air, in which case the same loss occurs; or it may be "drowned" —that is, it may be set below the water level in the tail-race; or it may discharge into a suction or draft tube. In these two latter cases the above-mentioned loss of head will not take place.

SUPPLEMENTARY DEVICES AND MANUFACTURED TYPES

ACCESSORIES

94. Diffuser. An apparatus for the purpose of providing a gradual enlargement of section for the passage of the discharge water as it leaves the runner buckets of a radial outward-flow turbine is called a *diffuser*, a device no longer in general use. It usually consists of two fixed conical surfaces flaring out opposite the discharge edges of the turbine crowns, giving a divergent profile to the walls of the passageway. The object of the diffuser is to prevent part of the loss of energy incurred in the general case, due to the residual velocity of the escaping water. In this case the absolute velocity of discharge at the extremities of the runner buckets is slowly

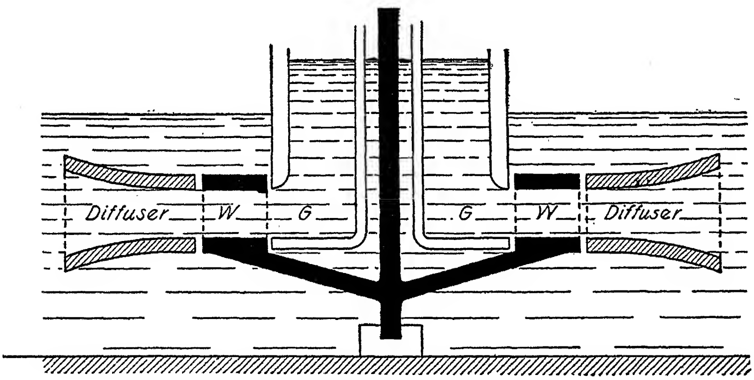


Fig. 84. Illustrating Operation of Diffuser

decreased as the cross-section of the passageway through the diffuser increases; and the discharge water finally passes out at a much lower absolute velocity than that at the runner, with a consequent gain in efficiency, see Fig. 84, and also Fig. 74. The efficiency of a reaction turbine is increased by making the exit depth d_1 greater than the entrance depth d, as shown by Equation (**109**); the stationary diffuser produces the same result.

95. Draft Tube. A wheel so set as to discharge above tail-race level loses some available head. If the discharge, however, takes place through a pipe filled with water, the lower edge of which is below the surface of the tailwater, most of this head becomes effective, the draft tube virtually adding this additional head to

the static head. In order that the tube may remain full of water it must be placed not more than 25 feet (the practical suction limit) above tailwater level. In practice draft tubes seldom exceed about 18 feet in length vertically, their main functions being to render the turbine readily accessible for examination and repairs, and to avoid the loss of head resulting from a discharge above tailwater level.

The draft tube is essentially a water barometer; hence, if v represents the velocity in feet per second of the water in the tube, the water column cannot be greater than $34-\frac{v^2}{2g}$, and any height of tube above this will result in a vacuum, involving a corresponding loss of head. In addition there must be considered losses due to entrance, friction, etc., as well as that represented by the velocity of the discharge water, the effect of which is to still further reduce the effective head on the wheel. It should be noted, however, that the water as it escapes from the turbine should have a velocity of not less than 2 or 3 feet per second in order to prevent air bubbles from rising and displacing the water in the tube, and also in order to carry off any air that might be contained in the tube at the time of starting.

To seal the draft tube against the passage of air beneath the lower tip, the latter should dip below the water surface to the extent of 6 to 24 inches, depending on the dimensions of the draft tube, the larger sizes requiring the greater depths, at the same time in every case insuring against the uncovering of the lower edge at low tailwater level. It will be seen later that the tube should be gradually increased in cross-sectional area and the lower portion flared out in trumpet shape. These tubes are usually of circular cross-section, and built of riveted steel, though recent practice under some conditions favors moulding them in concrete, occasionally lined with steel. From the nature of their action air-tightness is a prime requisite for efficient work.

Illustrative Example. A turbine, Fig. 85, receives a uniform supply of 20 cubic feet of water per second from a steel penstock 2 feet in diameter and 2000 feet long. The total drop from headrace to tailrace is 80 feet. The turbine is installed 10 feet above tailwater. Discuss its operation.

The mean velocity of flow in the pipe is

$$v=\frac{q}{\pi r^2}=\frac{20}{3.1416\times(1)^2}=6.4 \text{ ft. per sec.}$$

The velocity head of the moving water $\left(\frac{v^2}{2g}\right)$, is

$$\frac{v^2}{2g}=0.64 \text{ ft.}$$

The head lost in friction, taking the value of c as 105 (see "Hydraulics," Articles 28, 36), is

$$h_f=\frac{8gl}{c^2}\frac{v^2}{2g}$$

$$h_f=16.38 \text{ ft. } (FD \text{ in Fig. 85})$$

The head lost at entrance $\left(\text{taking } m=0.5, \text{ from } h_e=m\frac{v^2}{2g}\right)$, is

$$h_e=0.32 \text{ ft.}$$

In the figure, $CF=0.34+0.68=1.02$ feet; and $KD=70-(16.38+0.96)=52.66$ feet. Consequently the total effective energy deliv-

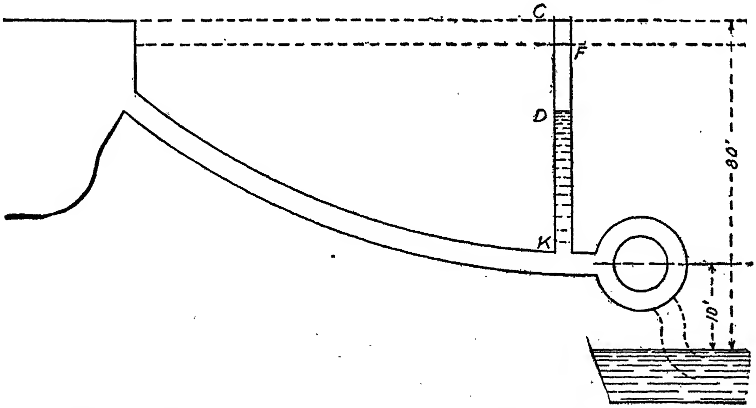


Fig. 85. Diagrammatic Representation of Turbine Installed, Showing Penstock, Headrace, and Tailrace

ered to the turbine is represented by 52.66 feet of pressure and 0.64 foot of kinetic energy, totaling 53.30 feet; but it has an additional gravity head of 10 feet chargeable to it, which is wasted if the motor discharges into the atmosphere, and practically all utilized if it discharges into a draft tube. In either case, the head h chargeable to the installed motor, and to be used in computations of

efficiency, is $53.30+10=63.30$ feet. At first sight it might appear that the head chargeable to the motor should be 80 feet, the total fall; but from what has preceded, it is evident that $16.38+0.32=16.70$ feet head is utilized in overcoming penstock resistances, and is not properly chargeable against the motor, either as a separate machine, or as installed. In the former case, the head h should be 53.30 feet; in the latter, 63.30 feet.

Thus with a diameter of 2 feet, the loss of head in the penstock is $16.38+0.32=16.70$ feet; and the power lost is $16.70\times20\times62.5=22{,}545$ foot-pounds per second $=40.99$ horsepower.

If the turbine discharges into the atmosphere, the head of 10 feet is also lost; this would occasion a further loss of $10\times20\times62.5=12{,}500$ foot-pounds per second, or 22.7 horsepower. With a draft tube, this last loss would be avoided.

Supplementary Example. Suppose a 3-foot penstock to be substituted for the 2-foot, the discharge remaining as before.

The mean velocity in this case is 2.84 feet per second.

The head lost in friction (supposing c to be 110 in this case) is

$$h_f=1.8 \text{ ft.}$$

The head lost at entrance is

$$h_e=0.06 \text{ ft.}$$

The velocity-head is

$$\frac{v^2}{2g}=0.12 \text{ ft.}$$

Thus the loss of head in the penstock is $1.8+0.06=1.86$ feet, which represents a loss of $1.86\times20\times62.5=2325$ foot-pounds per second, or 4.2 horsepower.

Flaring or Conical Tube. When draft tubes are used, they should be of the flaring type, so as to change the speed of the water gradually as previously explained. This largely prevents the loss of energy in shock which always results when water, issuing from the runner at relatively high speed, impinges on the mass of slower moving water in the draft tube. It has already been shown that the head corresponding to the absolute velocity with which the water leaves the wheel is, in the general case, entirely lost; but when the discharge takes place into a draft tube of conical or flaring shape (the cross-sectional area increasing regularly from the tur-

bine to the tailrace) the velocity of the flowing water is gradually reduced without shock or formation of eddies, as in the case of the diffuser. The change in cross-section of the tube should be made gradual in order that the corresponding change in velocity may be gradual; hence in certain conditions it may be judicious to increase the total length of the tube without increasing its vertical dimension by curving or inclining its axis, and this method of constructing the draft tube may sometimes result in a considerable reduction in the amount of excavation otherwise necessary for the tailrace.

Pulsations in the water under sudden changes of load often occur, especially when operating under a high head of water and with a quick acting governor. The tendency of these oscillations is to interfere with the speed regulation and in some cases the results may be disastrous. Experience seems to indicate that cylindrical draft tubes suffer more in this respect than conical; also the former less readily force out the contained air when the turbines are started, and do not hold the draft head under light loads as well as the latter.

Conservation of Discharge Head. *"The head corresponding to the velocity of discharge from the runner into the draft chest or tube may represent a considerable proportion of the total head acting on the turbine. This is especially true in the case of runners of high specific speed, where this velocity head may be as great as from 25 per cent to 30 per cent of the total head acting on the turbine. It is, therefore, essential to design the draft tube most carefully, in order to convert this velocity into effective head.

"Until the last two years a large majority of the turbines installed in connection with low and medium heads were of the old fashioned multirunner type, consisting of two, four, six, and sometimes more runners on a shaft, these runners being grouped in pairs, each pair discharging into a common draft chest and tube. The energy represented by the velocity at the discharge from the runners was practically lost to these turbines, owing to the losses due to impact and eddies and whirls existing in the draft chests and tubes as a result of the runners discharging against each other. Consequently, there was practically no reconversion of the velocity head from the runners into useful head.

*Abstract from a paper read before the Canadian Society of Civil Engineers, by H. B. Taylor, hydraulic engineer.

"Ten years ago the draft-chest and draft-tube designs of turbines of these types were exceedingly crude. Improvements have been made, of course, from time to time until the multirunner unit, when now built, is quite an efficient machine. The principal improvements have been in allowing a greater distance between the runners which discharge into a common draft chest, and in designing the draft chest so that the water from each runner is carried into the draft tube to a point where the two streams are moving in parallel directions, before allowing them to converge. But even with the best design of turbines of the multirunner type, there is

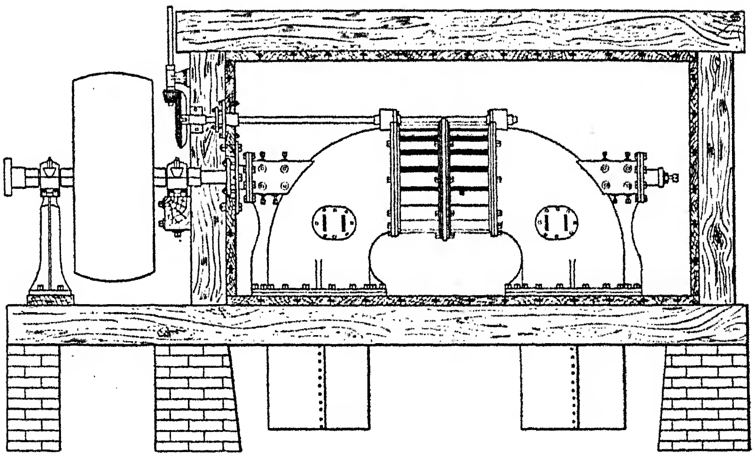


Fig. 86. Pair of Register-Gate Wheels in Wood Penstock, Each Discharging through an Independent Elbow into Its Own Draft Tube

always present an appreciable loss due to the comparatively sudden turns which the discharge water from the runners must necessarily follow.

"With runners of medium specific speed in a multirunner setting, it is sometimes possible to obtain efficiencies approaching those developed by the experimental runner; provided, of course, that the draft chests and draft tubes have been carefully designed. In other words, the losses in the draft chest and draft tube about equal the losses which exist at the Holyoke testing flume, where the runner is small, and where the intake- and draft-tube losses are necessarily appreciable on account of the construction of the flume. On the other hand, should runners of extremely high specific speed

be adopted, in which the discharge velocity from the runners represents a very considerable portion of the total head acting on the turbine, it is found that the efficiency seldom reaches that of the experimental model. It may be mentioned that the experimental model is usually operated in a vertical setting, with a single draft tube, and consequently the losses which exist in the draft chest of the multirunner setting are not present in the experimental setting."

Since the utility of draft tubes has been fully recognized and its employment become widespread, the use of turbines on horizontal shafts has become practicable, the connection being made by means of elbows or a tee, Figs. 86 and 87.

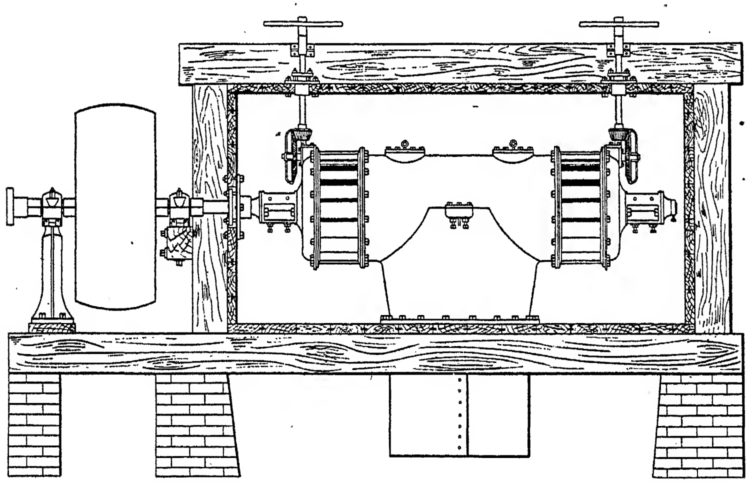


Fig. 87. Pair of Turbines in Wood Penstock, Discharging through Single Draft Tube

Although it is now true that draft tubes may be employed with more or less convenience in connection with any class of turbine, the Francis and the Jonval types and their modifications are most easily fitted with this device. When impulse wheels of the Pelton or other similar type are arranged to operate with draft tubes, it is essential that the upper surface of the water in the tube be maintained automatically below the lowest point of the revolving buckets, which then move in a partial vacuum in the top casing or head of the draft tube. Thus there is joined not only an additional draft head as already explained, but a decrease in the frictional air resistance for the rotating parts. It is clear, however,

that that portion of the draft head is necessarily lost which is represented by the distance between the water surface in the draft tube and the center of the nozzle-discharge opening; or, in case two or more nozzles are used, it is the mean vertical distance. In the case of an impulse turbine fitted with a draft tube in the manner described above, the head lost is the vertical distance, or the mean vertical distance, from the water level in the draft tube to the center of the guide-bucket discharge openings.

The principle of the air-admission valve for automatically maintaining a constant height of "hanging column" in the draft tube, is as follows: A vertical pipe at the side of the turbine has its upper end connected to the turbine case, and its lower end to the draft tube. In this pipe floats a copper ball connected with an air-admission valve in the turbine case; this float rises and falls with the water level, thus controlling the valve and the admission of air.

In European practice, gates are occasionally employed to close the lower end of draft tubes for the purpose of filling them with water before starting the turbines and for controlling the water supply to correspond with the varying load. But such gates become impracticable and expensive for large turbines, and their employment in practice to any great extent is improbable.

96. Fall Increaser. Under the name of *fall increaser*, Mr. Clemens Herschel has patented an apparatus designed to increase the fall acting on hydraulic turbines by the use of freshet water otherwise going to waste. In this way the normal output of power of a hydraulic power plant may be maintained at a constant normal quantity in spite of backwater, and for all those days in the year when there is water enough flowing in the river to produce so much as the normal output. In some cases of low fall, when there is an abundance of water to be had, also for certain cases of tide mills, the fall acting on the turbines may be increased throughout the year, over and above the natural fall, so as to produce a greater speed, and thus render the location more fit for generating electricity; while at the same time it will diminish the cost of the plant—generators, turbines, and building—per horsepower produced.

The fall increaser, shown underneath the turbines, Fig. 88, exhausts the turbine discharge from the vacuum box, and also produces a partial vacuum in this vacuum box, thus increasing the fall

that would otherwise act upon the turbines. At low water the fall increaser operating water is shut off, and the turbine discharge takes place under the natural fall, through the numerous holes in the casting, without material loss of head, as the velocity through these holes is only about 4 feet per second. Fig. 88 also shows diagrammatically the application of a siphon penstock in cases of very low water.

The value of fall increasers at any mill site depends on the measure of backwater liable to occur—that is, its rise and duration.

97. Regulating Gates. For many industrial purposes the power required of a turbine is variable, as when the number of machines operated in a factory is not constant, or when running dynamos to supply electric current to meet the fluctuating

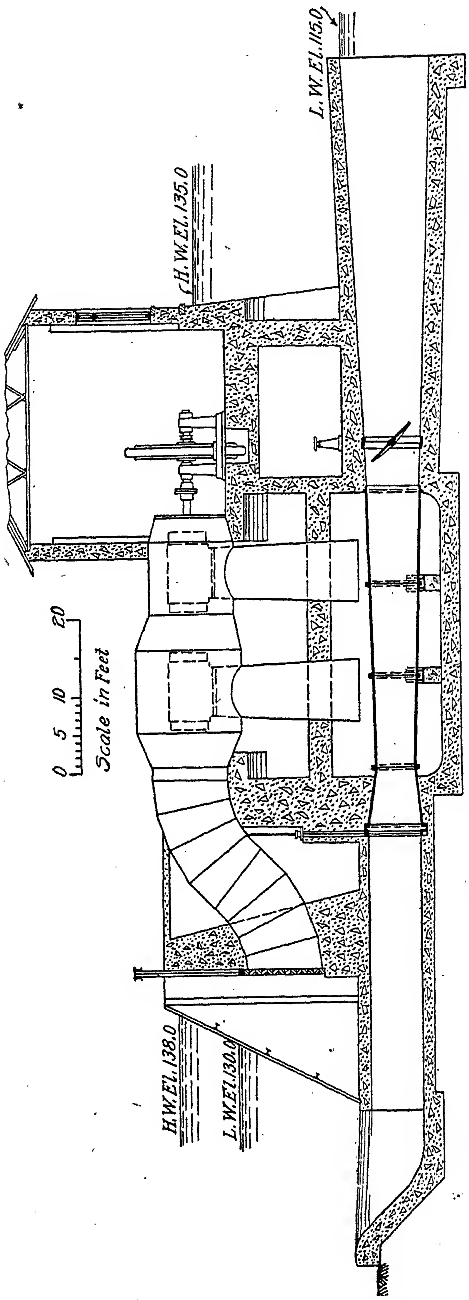


Fig. 88. Cross-Section of a 2400-Kw. Hydroelectric Plant with Herschel's Fall Increasers to Compensate for Backwater

demands of lighting or of transportation service; and since the speed of the turbine should be nearly constant, the quantity of water fed to the motors must vary. To do this without serious loss of efficiency at part and at full gate is a problem for the designer to solve. It may be said in general that with impulse turbines the regulation of feed water may be accomplished without serious loss by varying the cross-sectional area of the guide passages at the points of discharge, while with reaction turbines any attempt to regulate the flow by cutting down the feeding stream will entail more or less loss of energy by reason of the contraction and subsequent expansion of the stream which usually occurs.

A theoretically excellent method of regulation is that in which the water passages of guides and buckets increase and decrease

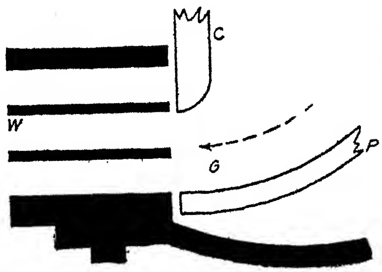


Fig. 89. Turbine Divided into Three Stories

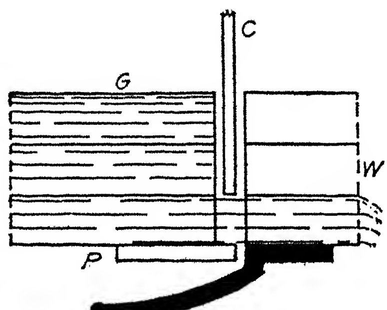


Fig. 90. Subdivision of Guide Passage

together, so that the roofs of the guide passages are always exactly in line with the crown of the turbine buckets, thus avoiding any change of section of the water stream. There are practical difficulties involved in such a design, however, and it is expensive.

Cutting down the flow by means of a gate in the penstock or draft tube results in considerable loss of energy; and to prevent such loss in the case of the plain cylindrical gate with axial motion, the turbine passages may be divided into stories by partitions, Figs. 89 and 90, as already described. The upper crown is sometimes curved downward for the same purpose.

Common Types. The three common types of gate are as follows:

(1) The *cylinder* form of gate is in most general use. It consists essentially of a cylinder placed on the inlet or outlet cir-

cumference of the runner, and by axial motion regulates the supply of water by blocking off to a greater or less extent the upper areas of the bucket orifices. As before stated, loss of energy will result when operating at part gate (partial closure), unless the width of the guide and runner buckets be partitioned off into two or more spaces, thus forming virtually two or more turbines with one regulating cylinder gate.

(2) The *register gate,* as the name implies, consists of a plate or cylinder with slots which correspond with the openings of the guide passages. The motion of the cylindrical type is circumferential about the common axis of turbine and gate, thus cutting off the areas of passage from the sides of the openings. In the case of parallel-flow turbines, the plate type of register gate must be used. In an ingenious modification of this type each guide vane is divided into two parts, a fixed and a movable section, all the latter being fastened to a rotating ring by means of which the clear openings can all be altered at the same time.

(3) In the *wicket* or *pivot* type of gate the guide vanes are pivoted in such a way as to afford an approximate balance at all openings. As in the case above, the simultaneous motion of all the vanes changes the openings of the bucket discharge uniformly.

The various forms of gates may be seen in many of the turbine illustrations shown in later portions of the text.

98. Case. This term includes the stationary part which supports the guides, gates, usually the gate operating devices, and the step on which the shaft runs, and it is customarily considered a component part of the wheel, as furnished by the maker. It usually includes two plates—one to hold the guides in place, the other to relieve the wheel of water pressure, though in some cases a single plate is designed to accomplish both purposes. Occasionally the term is used to designate the closed turbine chamber or casing which contains the wheel and appurtenances, and into which the feed water flows from the penstock.

99. Turbine Chamber or Flume. The turbine is placed in a vessel or chamber which receives a supply of water from the penstock or headrace and discharges through the turbine into a tailrace either directly or through a draft tube. This vessel is called an *open turbine chamber* or a *closed turbine chamber* (*casing*),

as the case may be. The term *flume* is occasionally used with a similar significance, though it should preferably be applied to a

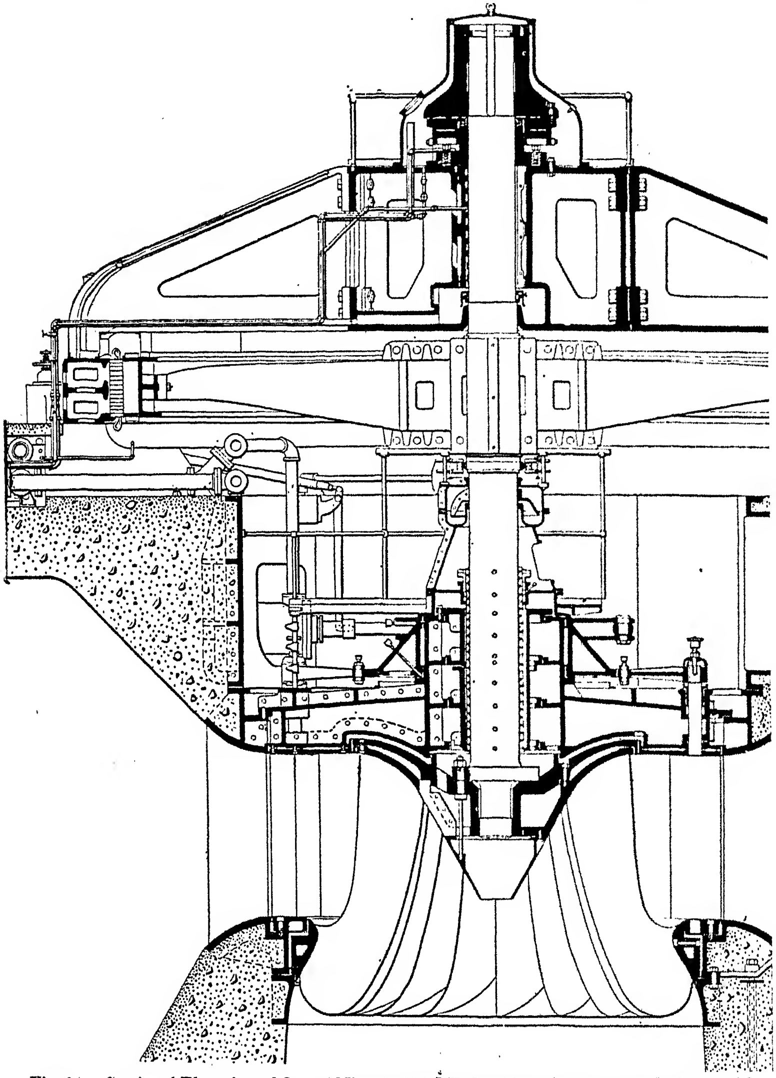


Fig. 91. Sectional Elevation of One of Nine 10,800-HP. Turbines Built for the Cedar Rapids Manufacturing and Power Company
Courtesy of I. P. Morris Company, Philadelphia, Pennsylvania

conductor of water not under pressure. The turbine chamber may be built of wood, steel, stone, concrete, Fig. 91, or some combination of these materials.

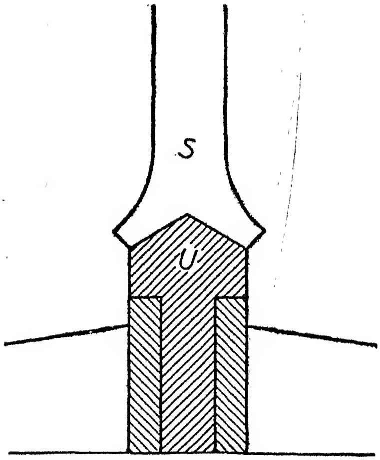


Fig. 92. Vertical Section of Lower Part of Wheel Shaft and Wooden Step Bearing

Open chambers have several advantages over closed: frictional resistance is lower; the wheel and wheel parts may be more easily reached for examination and repair; speed regulation is more readily achieved. In the case of open chambers, in order to avoid the suction of air into the turbines, a minimum depth of 4 to 5 feet of water should be maintained over the entrance rim of the guide buckets.

100. Bearings. The vertical shafts of turbines may be arranged to rotate on wooden step bearings, which, in some cases, are themselves free to turn in their sockets. The purpose of this arrangement is to permit the bearing to turn on its lower surface in case the frictional resistance of the upper surface increases to a material extent from heating or any other cause. Lignum vitae, maple, and oak are commonly employed; they are thoroughly dried, and then boiled in linseed oil until they become, in a measure, self-lubricating. Fig. 92 shows a vertical section of the lower part of a wheel shaft *S* and wooden step bearing *U*.

According to Thurso, "the use of wood with water lubrication for bearings and steps located under water has been practically abandoned by European manufacturers; and metal bearings and steps, with forced-oil lubrication, are employed instead, using a

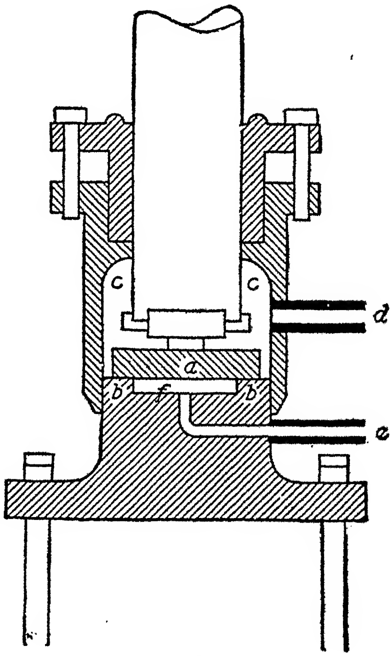


Fig. 93. Oil Step Bearing

pressure and return pipe for circulating the oil". Frizell shows a type of such oil step bearing, Fig. 93. The shaft passes through a stuffing box, and rests on the revolving plate *a;* the surfaces of contact between *a* and *b* are dressed to an exact fit, and wear keeps them in that condition. Oil for lubrication passes through the pipe *e,* being forced in under pressure by a pump; it fills the space *f,* and exerts a lifting pressure on the plate *a,* nearly equal to the weight of the shaft and its attachments, so that the contact surfaces sustain very moderate friction. The oil passes between these surfaces into the space *c,* and therefore the upward pressure on plate *a* would be neutralized, were it not that another pipe *d,* communicating with this space, conveys the oil back to the tank from which the pump draws its supply.

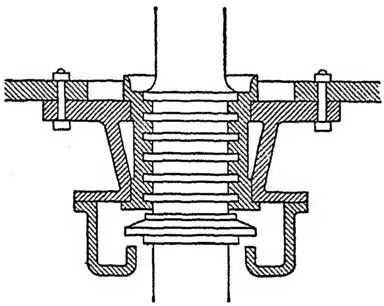


Fig. 94. Metal-Collar Thrust Bearing

A metal-collar thrust bearing is shown in Fig. 94. It is not advisable to place such metal bearings in the water, but on the end of the shaft opposite to that from which the power of the turbine is taken off. Thurso recommends that "both the straight and the collar bearings of the main turbine shaft should be adjustable, and should be lined with bronze as a base, and the bronze in turn lined with an antifriction metal, or babbitt, well hammered and bored".

*Guide Bearing.** "During the past two years, the lignum vitae guide bearing has come into general use with vertical turbines for both low- and high-head installations. The bearing now used must be distinguished from the old fashioned lignum vitae guide bearing commonly employed years ago, which consisted of three large lignum vitae blocks evenly spaced around the circumference of the shaft, each block being equipped with adjusting screws for taking up the wear. This design presented a small amount of bearing area to the shaft, a portion of each block immediately in line with the adjusting screw taking most of the load. The load

*From "Review of Recent Developments of Hydraulic Reaction Turbine", by H. B. Taylor, hydraulic engineer, *General Electric Review*, June, 1914.

on the bearing was, therefore, concentrated at the centers of the blocks, resulting in rapid wear and making frequent adjustment and renewal necessary.

"The bearing now used is so designed as to present a somewhat greater amount of projected area to the shaft than first-class practice would call for in the case of a babbitted bearing. The lignum vitae is dovetailed into the bearing boxes, Fig. 91, in the form of strips running parallel to the axis of the shaft and with the end grain of the wood placed normally to the surface of the shaft. Twenty or more of these strips, evenly spaced in a liberal length and separated by spaces for the circulation of cooling water, are so proportioned as to present sufficient area to the shaft to insure very satisfactory performance. The resulting bearing pressure per square inch may be made so light as to eliminate the provision of any adjustment for taking up the slight amount of wear which may take place.

"In the case of turbines operated in clear water, the supply for the bearing may be taken through a pipe directly from the wheel casing. A duplex strainer is connected in the line to remove any foreign substances which might otherwise reach the bearing and damage it. In installations in which the water carries large quantities of foreign matter in suspension, the power companies arrange a suitable central filtering system from which the filtered water is piped to the bearings of each turbine.

"The chief advantages secured by the use of the lignum vitae bearing as compared with the ordinary babbitt type of oil bearing which has often been used for a turbine guide bearing, are as follows:

"(1) The possibility of locating the bearing much closer to the runner, thus avoiding a considerable space by which the runner must be overhung below the bearing when the babbitt bearing is used in order to accommodate an oil shedder, an oil catcher, and seal ring.

"(2) No pump is required to remove the lubricant from below the bearing, as in the case of an oil bearing, and to pump the oil back into the lubricating system at a higher elevation.

"The chamber below the lignum vitae bearing is vented through cored passages in the runner hub into the draft tube, and it is into the draft tube that the circulating water from the lignum vitae

bearing escapes. Difficulties have been experienced in the operation of babbitted oil bearings located at this point, on account of the difficulty of providing a seal below the oil catcher which will satisfactorily prevent escape of oil into the draft tube and the leakage of water into the oil catcher, both of which actions involve loss or inconvenience. It should also be considered that the necessity for an oil circulating pump required by the babbitt type of bearing and the dependence which must be placed on this pump is an element of weakness, since any interruption to the lubricating system endangers the burning out of the bearing and the shutdown of the unit."

Thrust Bearing. "The thrust bearing of vertical wheels is now almost universally located above the generator on a cast-iron supporting truss which forms at the same time the generator-head cover, Fig. 91. The location of the thrust bearing above the generator has two advantages; *first,* it places the thrust bearing in the most convenient and accessible location for dismantling and inspection; and *second,* it eliminates the requirement of a thrust deck in the substructure of the power house, thereby reducing the cost of the power-house substructure. The thrust-bearing support is sometimes included with the generator and in some instances it has been constructed by the turbine builders as a part of the turbine machinery.

"The hesitation with which engineers formerly adopted the vertical type of unit and the preference which some engineers still express for horizontal-shaft machines, have been due almost entirely to doubt concerning the reliability of the thrust bearing. Today there are, however, at least two types of bearings of demonstrated reliability which are giving satisfactory service in this application.

"The old fashioned pressure thrust bearing, in which oil under high pressure is pumped into an annular chamber between a revolving disk and a stationary disk, was used almost exclusively in large units until about 5 years ago, when it was generally superseded by the roller bearing or a combination of the roller and pressure bearings.

"Since 1912 the Kingsbury type of bearing has come into general use and has given eminently satisfactory results through a wide range of conditions. The roller type of bearing has also a number

of important advantages and has given entire satisfaction in a number of notable installations."

Pressure Thrust Bearing. "In comparing the various types of bearings, it may be stated that the pressure thrust bearing when taken in connection with the necessary pumps and auxiliaries is the most expensive type both with respect to first cost and maintenance, and also in waste of power. An excessive dropping off of pressure or a momentary failure of the oil supply to the bearing will result in its immediate destruction.

"When most successfully used, the oil-pressure type of bearing has been operated on the principle of an oil supply of constant quantity. The portion of the thrust-bearing load which is due to the hydraulic thrust of the turbine runner increases with the gate opening of the turbine, thus requiring an increase of pressure in the thrust bearing corresponding to the increase of load. As the pressure increases, it is essential that the quantity of oil supplied to the bearing shall not be diminished, and this can be best accomplished by means of a positive-displacement type of pump such as a triplex-plunger pump geared directly to the turbine shaft. In starting up, a motor driven pump supplies the oil until the unit has been brought up to speed, after which the geared pump is thrown in.

"The interconnection of the thrust-bearing and governor systems by the use of one set of pumps for both, is dangerous practice, as is also the supplying of the bearings of the various units in the station from a single pressure main. When the governor system is interconnected with the thrust bearings, since there is no connection between the demand on the pump by the governor and the amount of oil required by the thrust bearing, any increase in the load on the thrust bearing may result in a drop in the quantity of oil supplied to the bearing, oil being diverted to the governor system in the line of least resistance. This action need not be carried very far to cause the failure of the bearing.

"In some instances where the governor and thrust-bearing systems have been interconnected, the attempted solution of the problem was by means of providing a pressure far in excess of that required for normal conditions in either system, the idea being that even should the governor tend to rob the thrust bearing of its proper

share of the oil, the resultant drop in pressure will still be insufficient to reduce the thrust-bearing pressure to an unsafe value. This method has many objections.

"The provision of an accumulator in the thrust-bearing system in an attempt to create a steady supply to the bearing is not considered good practice, since the cushioning effect in the accumulator is opposed to the above principle of constant quantity supplied to the bearing. If the supply to the bearing can be cushioned by compressed air in an accumulator tank, there is danger of the thrust disks coming together and burning out under such conditions.

"Most of the failures which have taken place in pressure bearings can be attributed to some condition in the pressure system which results in a variation in the rate of oil supply to the bearing. In brief, therefore, the pressure thrust bearing requires a separate pump for each unit and each pump of the positive-displacement type, giving constant oil quantity with pressure varying automatically to suit the thrust."

Roller Bearing. "The roller bearing was introduced somewhat in advance of the Kingsbury bearing. A distinct point in favor of the roller type of bearing is the fact, which is borne out by many experiences during operation, that even should the bearing be injured by defects in the rollers or tread plates, the bearing will still operate over long periods of time even in the defective condition without allowing the rotating parts to drop from their position of alignment.

"It is believed that the minor difficulties which developed in first starting up these bearings in the Keokuk plant were due largely to the use of a combination of the roller and pressure bearings, which combination appeared to introduce complications in the operation of the roller bearing. The purpose of the combination bearing is, of course, to furnish an alternative method of carrying the load, as a provision against failure of either type.

"Regarding the life of very large roller bearings, it may be said that in one of the early units installed in the McCall Ferry plant the bearing had been in continuous satisfactory service for over 3 years at the time the plates were reversed; and after a similar period the use may be continued by a slight regrinding of the plates.

"It is characteristic of the roller bearing that a bearing designed for a certain speed and load will run even more satisfactorily at lower speeds and is less suited to prolonged operation at higher speeds. The Kingsbury bearing, on the other hand, operates even better at high speeds than under normal conditions, but sometimes develops trouble if run too long at an extremely low speed. Thus, any trouble which may develop in these bearings is likely to appear in the roller bearings at runaway speed of the unit and in the Kingsbury bearing when starting or stopping the unit."

Kingsbury Bearing. "Referring again to the McCall Ferry plant, which seems to furnish the best experience in actual service of large thrust bearings, the writer has been advised that the operation of all the Kingsbury bearings has been entirely satisfactory. The first Kingsbury bearing was put into service in 1912, and frequent examinations have indicated that the wear is so small as to be negligible and the life of the bearing, excluding accidents, is estimated at over 25 years without rebabbitting the bearing shoes.

"In the Kingsbury bearings installed at Keokuk, trouble was experienced in some of the first bearings put into operation, due to a "wiping" of the babbitt at the instant of starting the unit from rest. In the few cases where wiping has developed, it has been due either to lack of proper finish of the rubbing faces of the collar or to the presence of air in the oil. Such troubles as were experienced during the early operation of the Kingsbury bearings at Keokuk have disappeared so that all of the bearings have now been in operation for 8 months or more without giving trouble.

"An interesting combination of the two types of bearing will be installed in the plant now under construction at Cedar Rapids, Canada, in which the thrust of each unit is carried normally by a Kingsbury bearing, Fig. 91. A roller bearing of reduced dimensions is placed within the Kingsbury bearing, but is allowed normally to remain out of action by the provision of a slight clearance. Should any wiping action take place in the Kingsbury bearing, the rotating parts would then settle a slight distance and the load be transferred to the roller bearing. While the roller bearing is not sufficiently large to carry the load indefinitely, it is of sufficient capacity to operate long enough for the station operators to prepare for a shut-down of the unit, thus providing against any injury to the turbine."

101. Thrust Compensators. *Balancing Piston.* This device is principally used for horizontal inward- and outward-flow turbines to take the thrust both of the action of the water and the weight of the rotating parts. Fig. 95 (Niagara Falls Power Company, Power House No. 1*) represents an example of twin-flow wheels on the same shaft. The weight of the dynamo, shaft, and turbine (about 70 tons) is balanced, when the wheels are in motion, by the upward

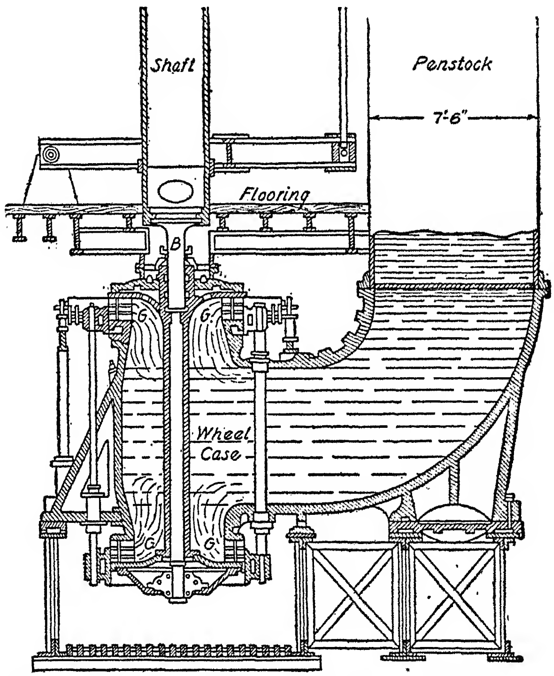


Fig. 95. Thrust Piston with Twin-Flow Wheels Mounted on the Same Shaft in Niagara Falls Power Company's Power House No. 1*

pressure of the water in the wheel case on a balancing piston or disk *B*, placed above the upper wheel and rigidly attached to the shaft. The upper disk containing the guides is perforated so that the water pressure existing in the penstock can be transmitted directly through to the lower side of the balancing piston, while the upper side is open to atmospheric pressure. The lower disk is not perforated; and the weight of the water upon it is carried by inclined rods upward to the wheel case, which, together with the

* In the original installation. See Part III.

penstock, is supported upon several girders. At the upper end of the shaft is a special thrust bearing designed to receive the excess of vertical pressure, which may act either upward or downward, under different conditions of power and speed.

Fig. 96 shows diagrammatically the device applied to their single inflow turbines by the Niagara Falls Power Company, in Power House No. 2. The water from the penstock fills the annular chamber *A* under nearly hydrostatic pressure; passes through the guide passages at *G;* and enters the wheel channels at *C* under reduced pressure and at high velocity. The revolving turbines, shaft, and attachments are shown in black. The water, leaving the turbine channels *W*, enters the space *D* with low absolute velocity and low pressure. At the lower end of the shaft, while lateral support is provided by the step bearing, a great lifting force is furnished by the admission of water directly from the headrace by an independent pipe under the full headrace pressure to the space *U U* on the under side of the conical shell or balancing disk *V V*, which is keyed upon the shaft and revolves with it. The pressure on the upper surface of the piston is small, being that of the water in the upper end of the draft tube. In this way the larger part of the weight of the wheel, shaft, and armature of the electric generator is supported by fluid friction. The diameter of the balancing disk is 4.9 feet; the weight of the revolving mass is 71 tons, of which 66 tons are supported by the upward pressure of the disk, leaving 5 tons to be sustained by a suspension or collar bearing at the upper end of the shaft.

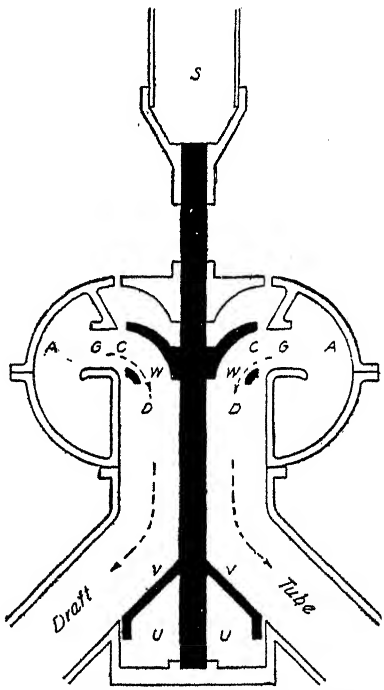


Fig. 96. Thrust-Piston Device for Single Inflow Turbine in Niagara Falls Power Company's Power House No. 2

Thrust Chamber. The function of this contrivance is to resist the end thrust of vertical inflow turbines. It consists of an annular chamber opening toward the runner, which revolves in front of it. The water pressure in the chamber is supplied by a pipe connected with the penstock, and provided with a valve by means of which the end thrust can be regulated.

In double turbines working on the same shaft, the end thrusts balance and neutralize each other, provided both work at the same gate opening. Thurso recommends the employment of thrust chambers for single turbines working under a high head, say several hundred feet; and in case "the runners are of such size or shape as to preclude the use of such chambers, the thrust piston should be used instead, placing the runner at one end of the case, and the piston at the other.

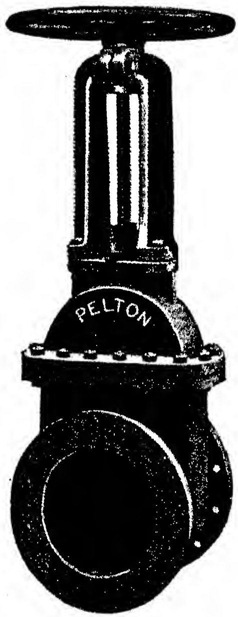


Fig. 97. Straightway Valve of Single-Disk Type, with Rising Spindle, Intended for Pressure on One Side Only. Disk rises clear of opening, giving a perfectly free way for the water. Seats are of bronze, to be replaced when worn.

"Turbines having thrust chambers or pistons, or double turbines so mounted that the two end thrusts balance, should nevertheless be provided with a small collar bearing, to take care of unavoidable variations in the end thrusts."

102. Valves. *Stop Valve.* For convenience of operation the supply pipe to each turbine should have a separate stop valve near the wheel for quickly stopping the turbine by shutting off its water supply. Other valves should be placed so as to divide the pipe line into convenient sections, any one of which may thus be cut out for inspection, repairs, renewals, etc. If the turbine is shut down by the closure of the regulating gates the turbine cannot be taken apart; moreover, under high heads the gates frequently allow sufficient leakage to keep the wheels in slow motion. Where several units are connected to the same penstock, separate stop valves, instead of a single head gate, are necessary to permit the shutting down of a single turbine for repairs.

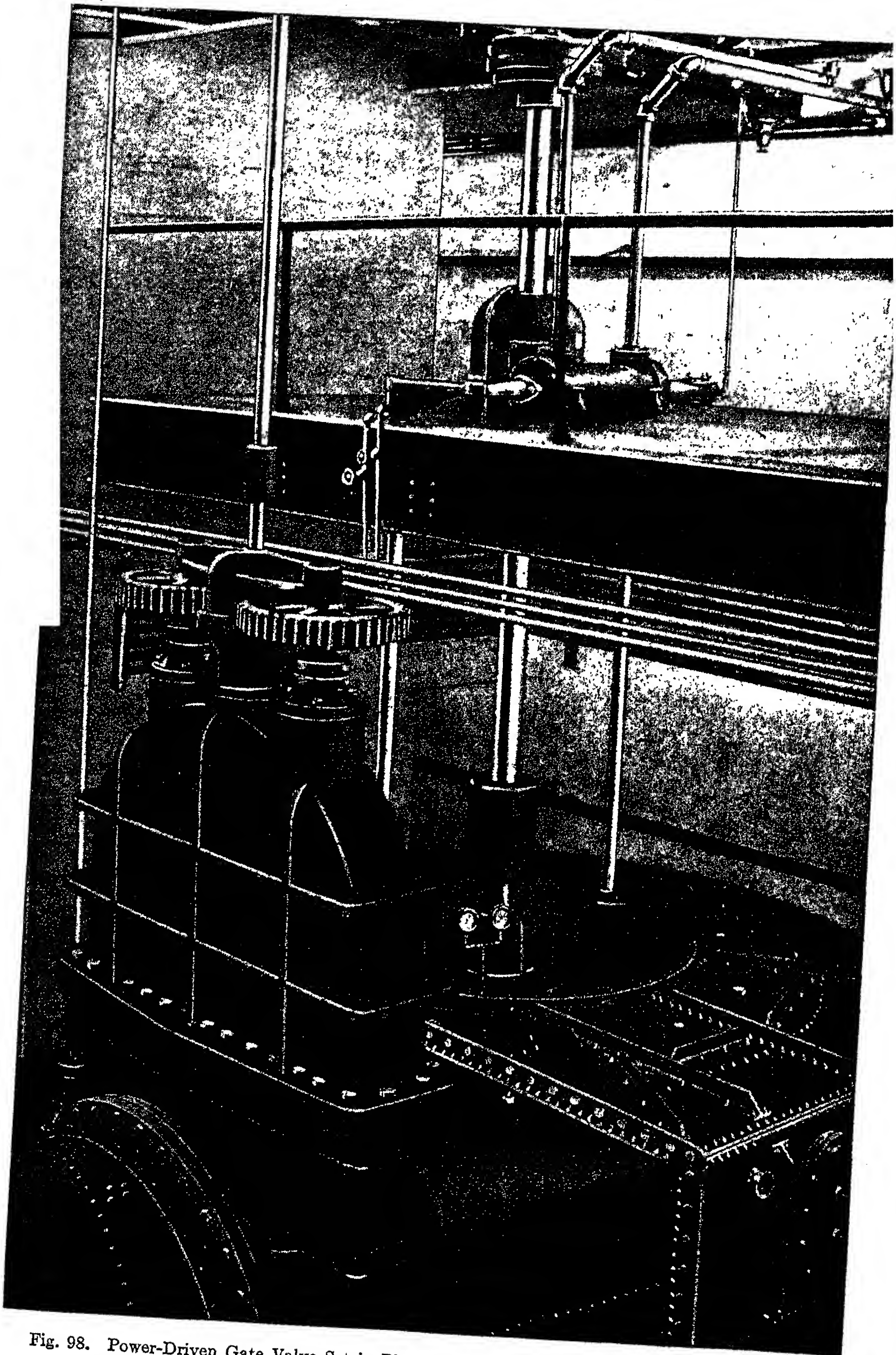


Fig. 98. Power-Driven Gate Valve Set in Plate Steel Spiral Casing, Leading to 900-H.P. Allis-Chalmers Vertical Turbine

In American practice gate valves are in common use, hand operated for the smaller sizes, Fig. 97, and power driven for the larger and heavier gates, Fig. 98. Electric, Fig. 99, hydraulic, Figs. 100 and 101, and pneumatic power have all been used. In the case of hydraulic power, where the head employed is high enough, the pressure water may be taken directly from the penstock; but for lower heads a pressure pump and weighted accumulator are required. Sometimes a by-pass is employed in connection with a gate valve for the purpose of partially equalizing or neutralizing the pressure, as shown in Fig. 102.

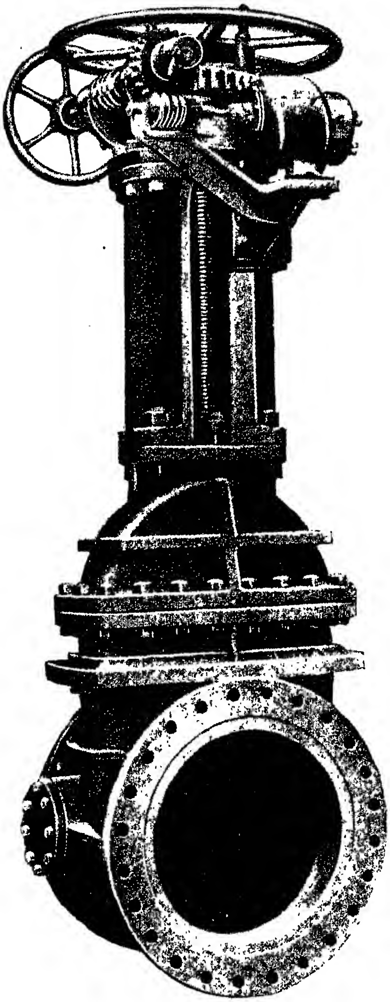


Fig. 99. 24-Inch Gate Valve of Single-Disk Type, with Outside Screw and Yoke and Rising Spindle. Arranged for operating by electric motor, and provided with roller bearings to take thrust from stem.

Courtesy of Pelton Water Wheel Company, San Francisco, California

Butterfly valves have a well defined sphere of usefulness in connection with certain phases of hydraulic work. They are not absolutely water-tight, as compared to gate valves, nevertheless the amount of leakage from a correctly designed and constructed butterfly valve may be so slight as to be readily discharged through a drain pipe. Butterfly valves are arranged for manual or electric-motor control, or for operation by a hydraulic piston.

Automatic Stop Valve. Automatic stop valves should be placed at critical points on the line, so that, in case of accident to the pipe, the valves will gradually close, and will thus prevent the loss of water and the possible damage to property.

Air Valve. At summits of a pipe line and near stop valves, air valves should be placed for the purpose of permitting the escape

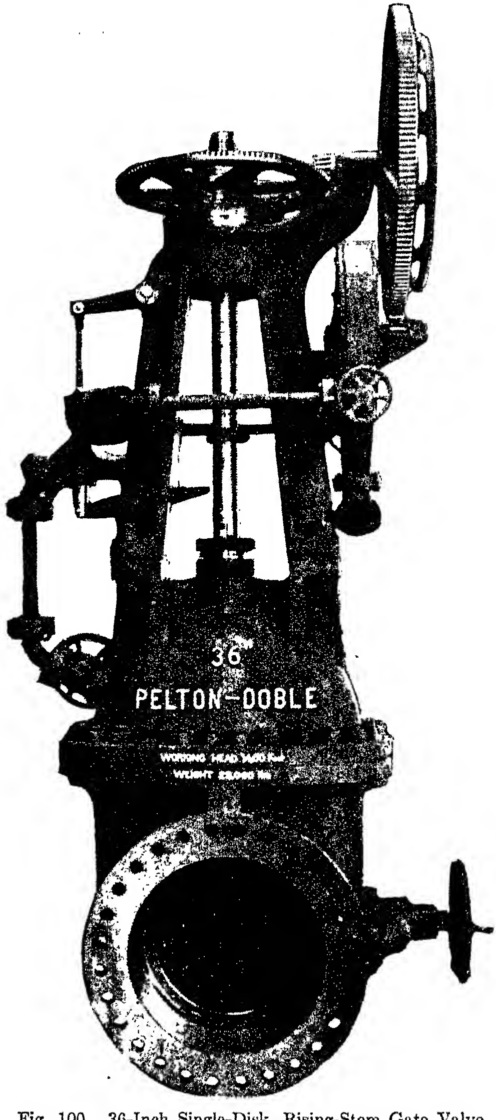


Fig. 100. 36-Inch Single-Disk, Rising-Stem Gate Valve for High Pressure. Operated by Reversible Water Motor
Courtesy of Pelton Water Wheel Company, San Francisco, California

of air in filling, the entrance of air on emptying, and occasionally

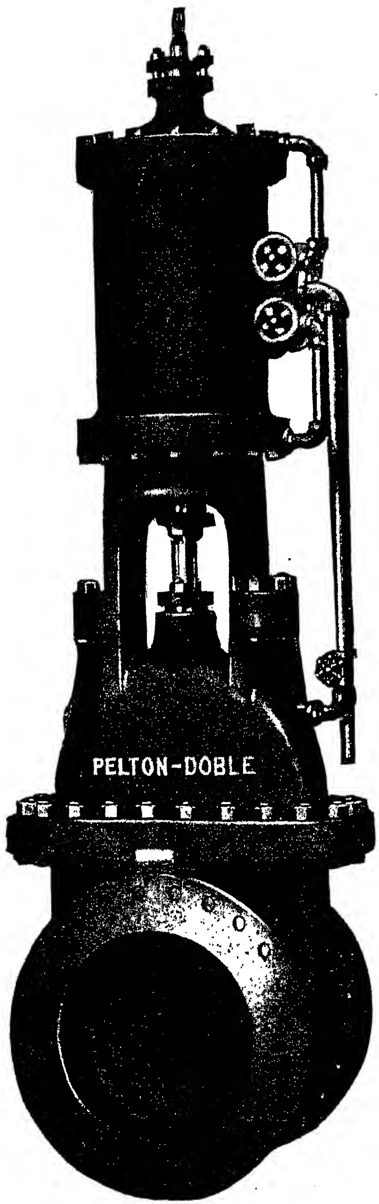


Fig. 101. 42-Inch Single-Disk, Hydraulic Gate with Operating Valve
Courtesy of Pelton Water Wheel Company, San Francisco, California

the escape of air which may gradually accumulate at summits. They are usually designed to operate automatically, Fig. 104.

Blow-Off Valve. These should be placed at all depressions of a pipe line, for the purpose of cleaning out or of emptying sections of the line.

103. Gages. For the purpose of readily ascertaining the character of the operation of a water-power plant it is necessary to install gages at definite points in the works—for example, a pressure gage near the entrance of the guide buckets; a vacuum gage near the discharge openings of the runner buckets; gages to indicate the pressure between the runner disk and the head of the case (or the dome), in the thrust chamber and behind the thrust piston when these devices are employed; also, at the end of a long penstock where water hammer and the fluctuations of pressure following speed regulation may be observed. The speed of the turbine itself should be read by a speed gage, while a dial and pointer may serve to indicate the gate opening, or position of the regulating gates.

104. Mechanical Connections. *Transmission from Shaft.* The simplest case is presented when the electric generators can

be placed directly upon and revolve with the turbine shafts as at the Niagara Power Company's installations; in such cases the revolving armatures may be made heavy enough to act as flywheels. In other cases the power may be transmitted to other shafts by means of spur or bevel gearing, or by belt or rope transmission—with considerable loss of power. (See Fig. 105.)

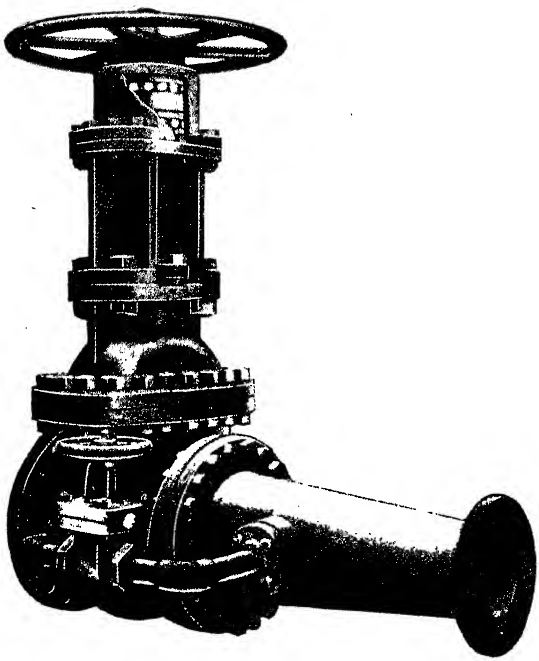


Fig. 102. Special High-Pressure Gate Valve with By-Pass, and Roller Bearings on Stem
Courtesy of Pelton Water Wheel Company, San Francisco, California

Engaging Mechanism. Mechanical devices for the purpose of throwing a wheel or shaft into or out of connection with the general system involve problems which differ in no essential particular from those met with in general mechanical-engineering practice, and therefore will not be considered here.

STANDARD MACHINES

105. James Leffel & Company. *Lower Head Types.* The types of turbines made by the Leffel Company, Springfield, Ohio, have long been in practical use. The type of runner, Fig. 106,

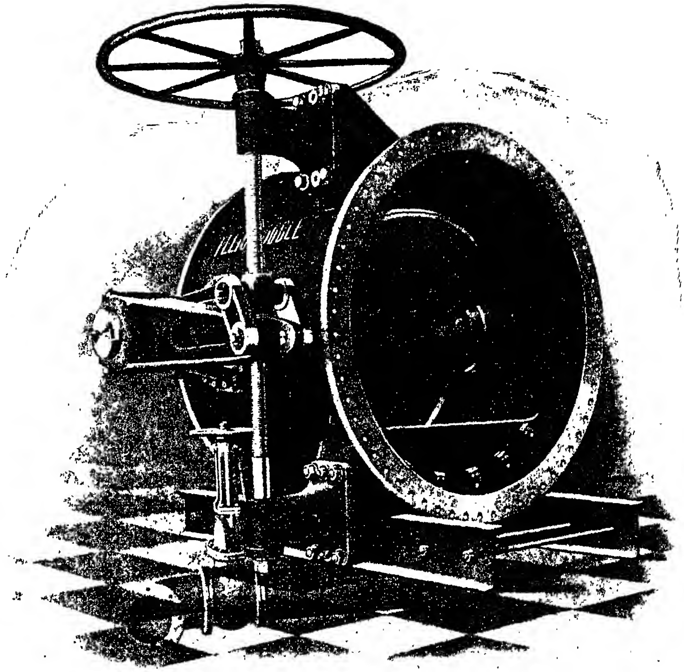


Fig. 103. 84-Inch Butterfly Valve Arranged for Hand Contro
Courtesy of Pelton Water Wheel Company, San Francisco, California

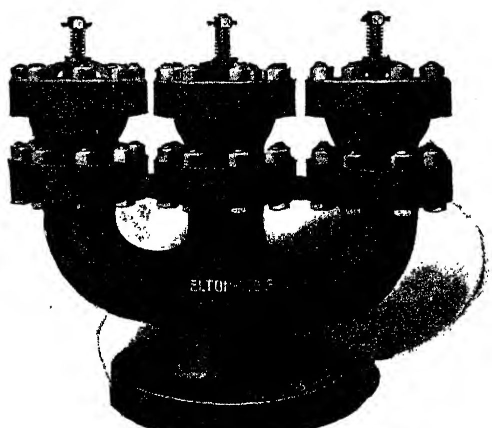


Fig. 104. Cluster of Pelton-Doble Automatic Air Valves

developed for the "Samson" turbines made by this company is a double wheel—the upper portion of the wheel passages being partitioned off by a diaphragm. Each wheel or set of buckets receives its separate quantity of water from one and the same set of guides, each portion of the water, however, acting on its own buckets. The complete turbine illustrated in Fig. 107 is arranged for vertical

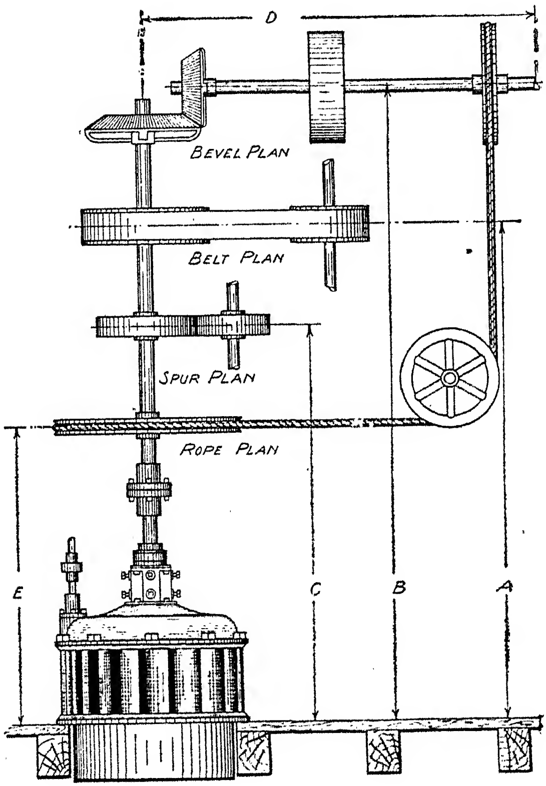


Fig. 105. Diagram Showing Various Methods for Transmitting Power

setting on the floor of a turbine chamber and shows the bearing and jaw coupling on the transmission shaft, as well as the wicket gates and gate operating mechanism.

In the simple type of wheel with quarter-turn draft chest, Fig. 108, the water discharges horizontally first and then vertically downward. The horizontal shaft and gate rod extend through stuffing boxes in one side of the turbine chamber—not shown—and

the horizontal iron base plate frequently is set directly upon its floor, usually with a draft tube extending below tailwater.

Two complete wheels on a horizontal shaft, one at each end of a plate-steel discharge case or draft chest, are represented in Fig. 109. Both wheels discharge horizontally toward each other, the water then passing downward through the central discharge pipe into a draft tube. The large iron base plate is set upon the floor of the turbine chamber, and the shaft may be extended in one

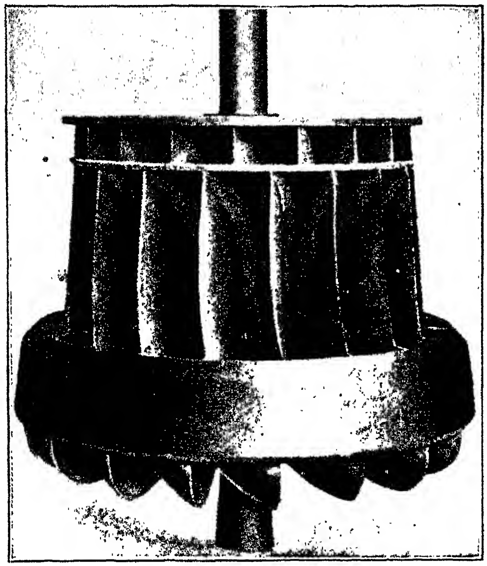


Fig. 106. Upright Shaft "Samson" Runner
Courtesy of James Leffel and Company, Springfield, Ohio

or both directions, passing through stuffing boxes in the sides of the turbine chamber. The small horizontal shaft on top of the cylinder has rigidly fastened to it two pinions so that by turning this shaft the regulation of both gates is effected simultaneously through the action of the rods.

Metal Casings for High Heads. The types represented in Figs. 108 and 109, are supposed to be placed on the floor of a simple open or decked turbine chamber of wood or other simple construction, when low heads are utilized. For the higher heads, particularly

when economy in space is necessary, there are built around the wheel iron or steel casings of sheet or cast metal.

In the cylindrical plate-steel casing, Fig. 110, the two wheels mounted on the same horizontal shaft discharge in opposite directions through curved elbows into the wheel pit or tailrace.

The Samson turbines are of the horizontal-shaft type, usually having one runner built with two similar sets of buckets taking

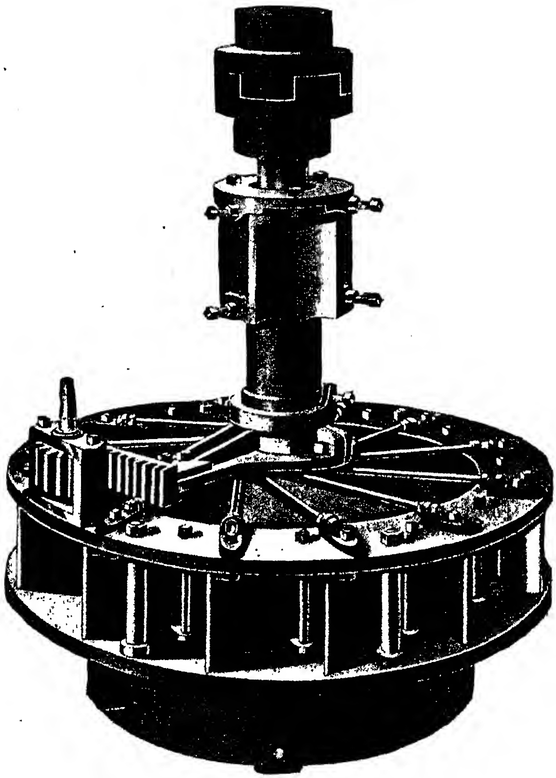


Fig. 107. Leffel Turbine without Globe Casing

the water equally divided from one set of guides and discharging in opposite directions.

106. S. Morgan Smith Company. The turbines built by the Smith Company of York, Pennsylvania, include the "Smith" types in many variations as well as the Francis type for the higher heads.

Smith-Type Turbines. The standard Smith turbine with wicket gates is shown in Fig. 111, set vertically in a plate-steel casing,

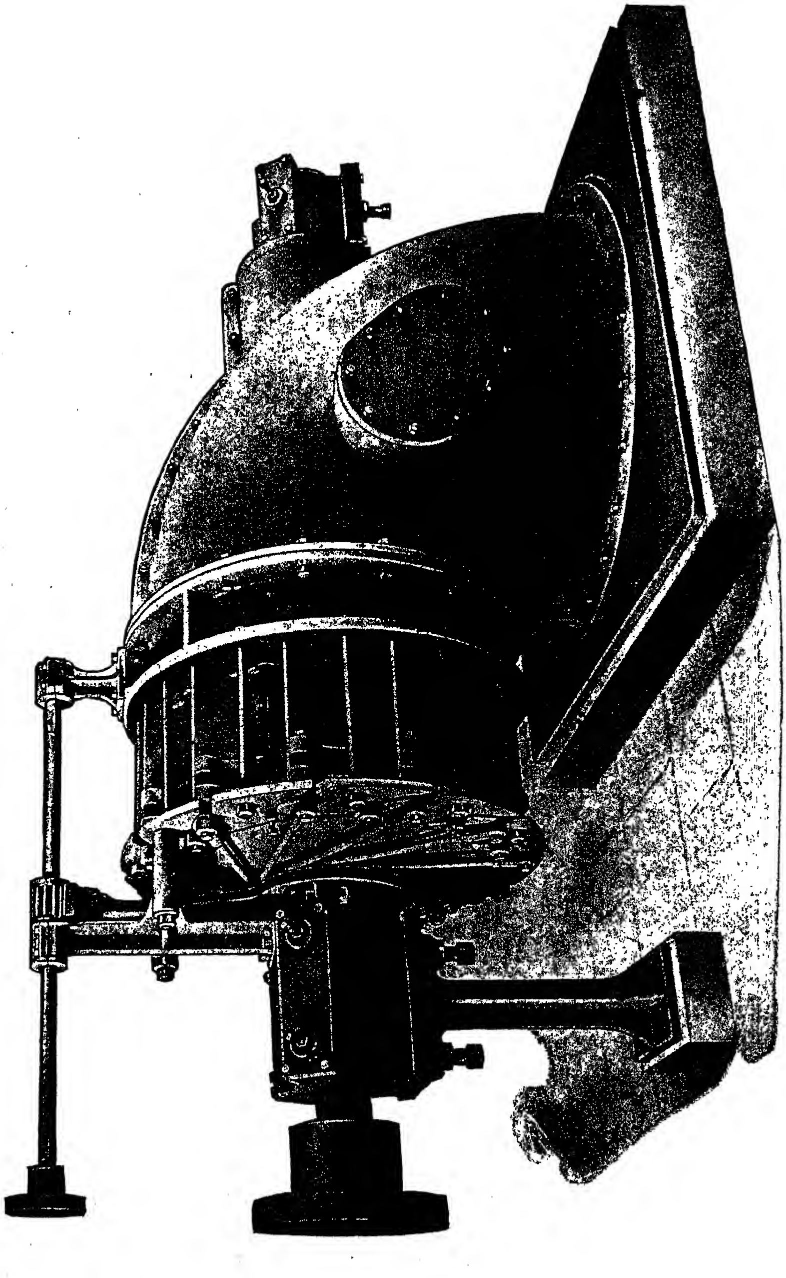


Fig. 108. Single-Discharge Horizontal-Shaft "Samson" Turbine for Open Penstock with Quarter-Turn Draft Tube
Courtesy of James Leffel and Company, Springfield, Ohio

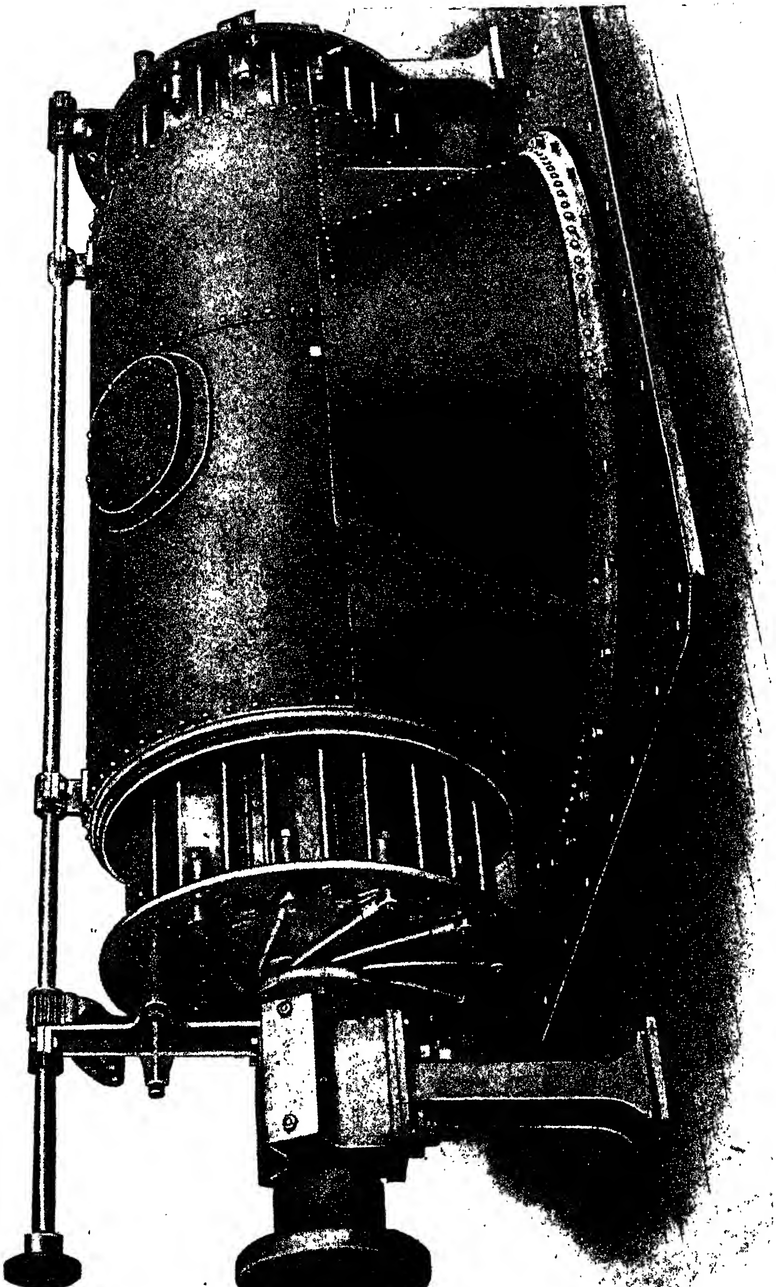


Fig. 169. Twin Center-Discharge Horizontal-Shaft "Samson" Turbines for Open Penstock
Courtesy of James Leffel and Company, Springfield, Ohio

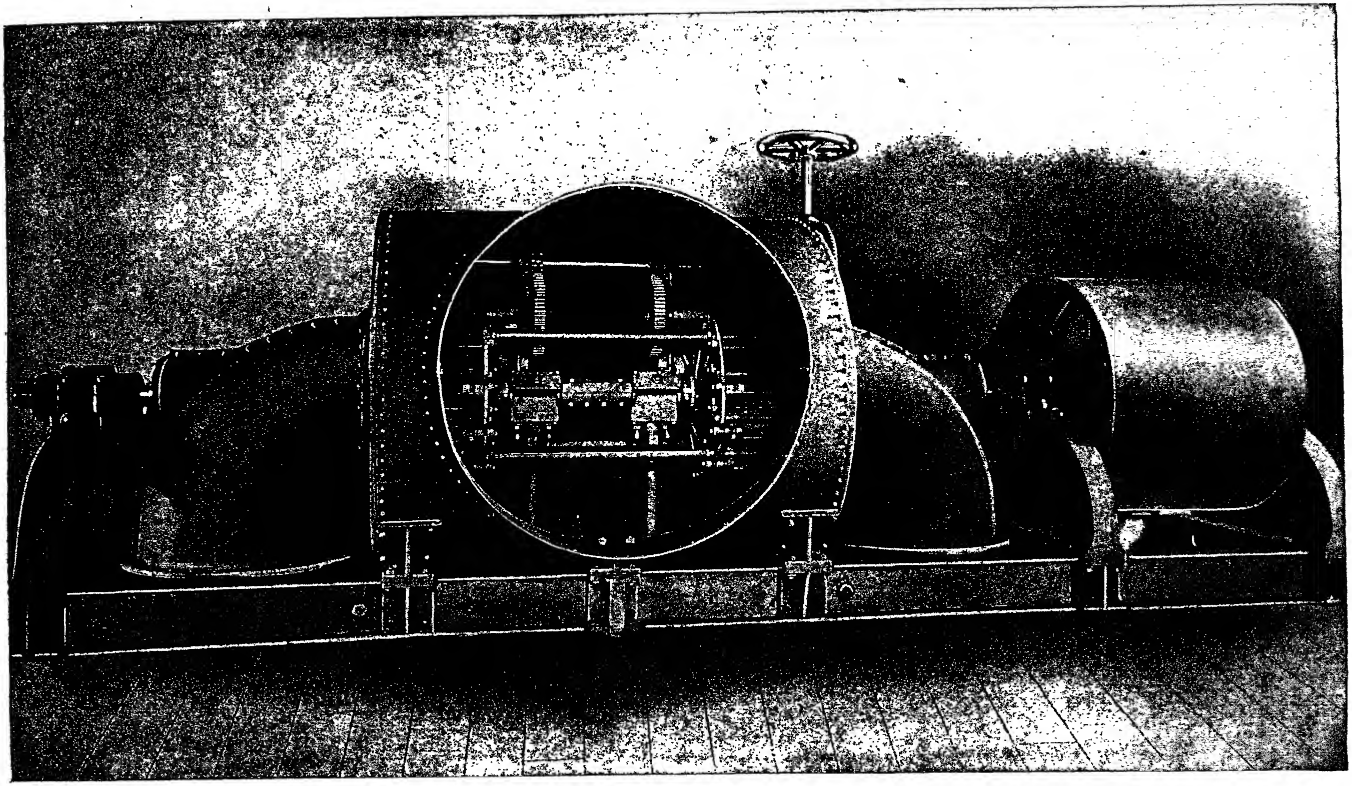


Fig. 110. Double-Discharge Horizontal-Shaft "Samson" Turbine with Casing
Courtesy of James Leffel and Company, Springfield, Ohio

and with bevel gearing arranged for belt drive. The same type of turbine fitted with a simple cylinder gate, designed to be easily removable without disarranging the operating mechanism, is illustrated in Fig. 112.

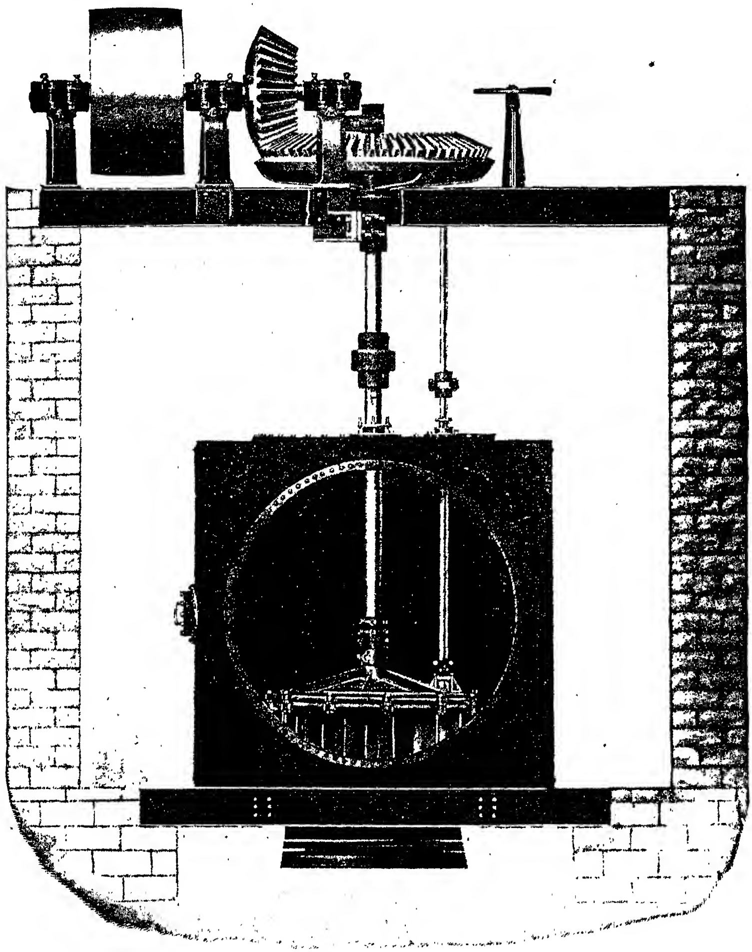


Fig. 111. Vertical-Shaft Turbine in Steel Flume
Courtesy of S. Morgan Smith Company, York, Pennsylvania

These machines are built to be set in open or closed flumes or enclosed in sheet-iron casings, and with shafts either horizontal or vertical. The vertical-shaft types may have spiral wheel casings of cast iron or formed in the concrete substructure. For the higher

power installations Smith turbines are arranged in multiple on the same shaft with draft chests and tubes to correspond, Fig. 113.

Francis-Type Turbines. Francis-type turbines are built with cast-metal spiral cases for high-pressure conditions, up to 600-foot head, and are arranged singly on vertical shafts, and singly or in pairs on horizontal shafts, with single or double discharge.

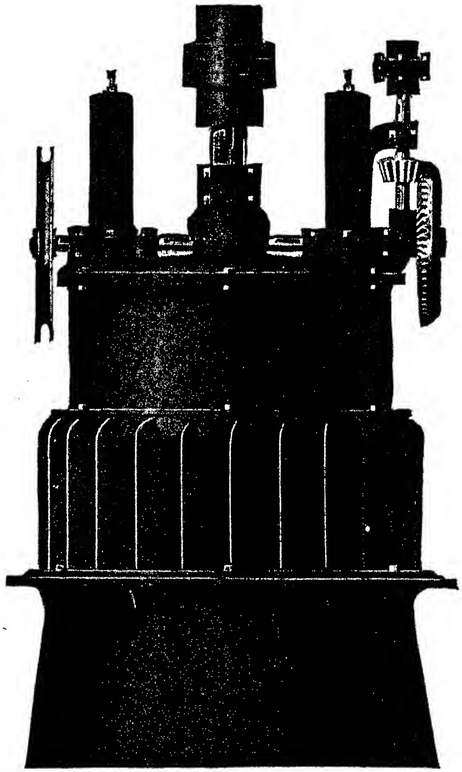


Fig. 112. Smith Turbine with Cylindrical Regulating Gates Fully Raised
Courtesy of S. Morgan Smith Company, York, Pennsylvania

107. Pelton System. The hydraulic machinery made by the Pelton Water Wheel Company, San Francisco, combining the patents of the former Abner Doble Company, includes Pelton-Doble tangential wheels and two general classes of Francis-type turbines—those with horizontal shafts and those with vertical shafts. Pelton equipment is standardized as to type of design only and is varied in certain elements to meet the particular operating requirements.

Tangential Water Wheels. Pelton-Doble tangential water wheels are built in standard and special designs of $\frac{1}{4}$- to 18,000- or more units.

The *Standard* Pelton wheel is ordinarily for timber-frame mounting and is used with either single or double nozzles. This wheel is supplied in nine sizes, the 6-inch to 24-inch diameters for heads up to 500 feet, and the other sizes up to 6-foot diameter for heads as high as 1000 feet. The use of the double nozzle may double the power with the same wheel, Fig. 114.

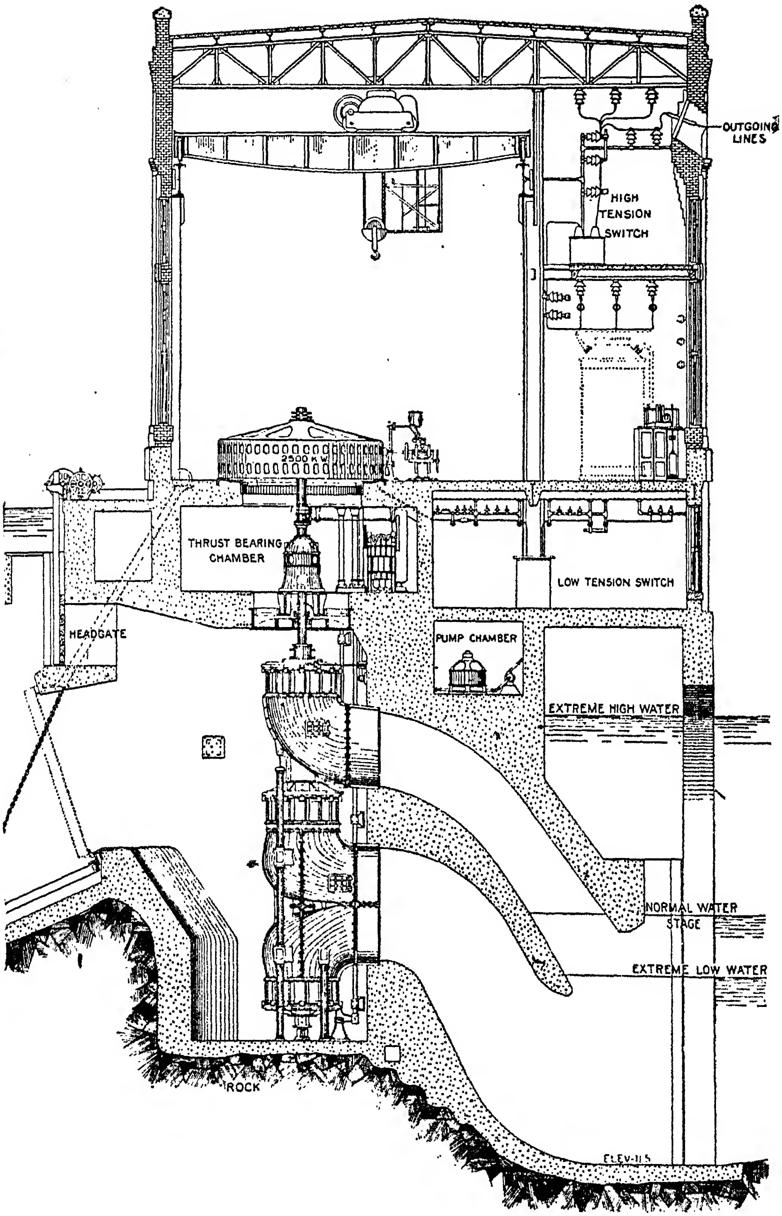


Fig. 113. Section of Installation of Smith Turbines Arranged in Multiple
Courtesy of S. Morgan Smith Company, York, Pennsylvania

Quintex nozzle Pelton wheels are operated with a nozzle having five openings controlled by a special integral gate valve and are

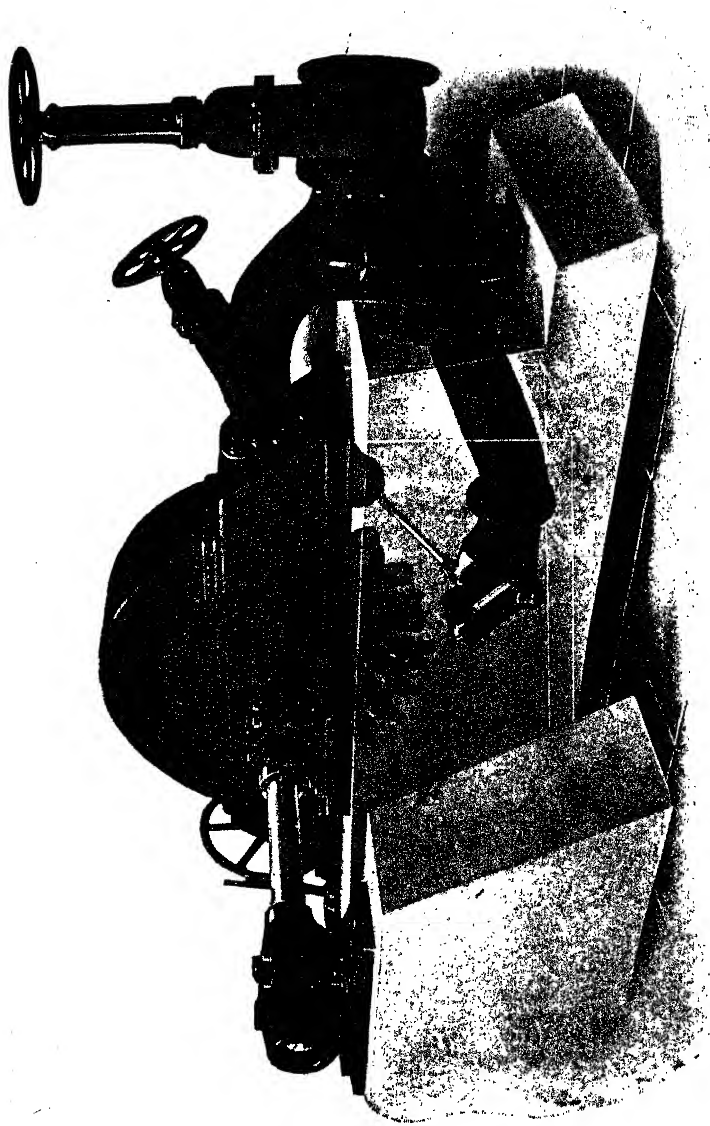


Fig. 114. Double-Nozzle Encased Pelton Wheel Equipped with Jet Deflector

practicable where this peculiar regulation is sufficient—for the lower heads, up to 50 feet, and for constant loads. They are built in five

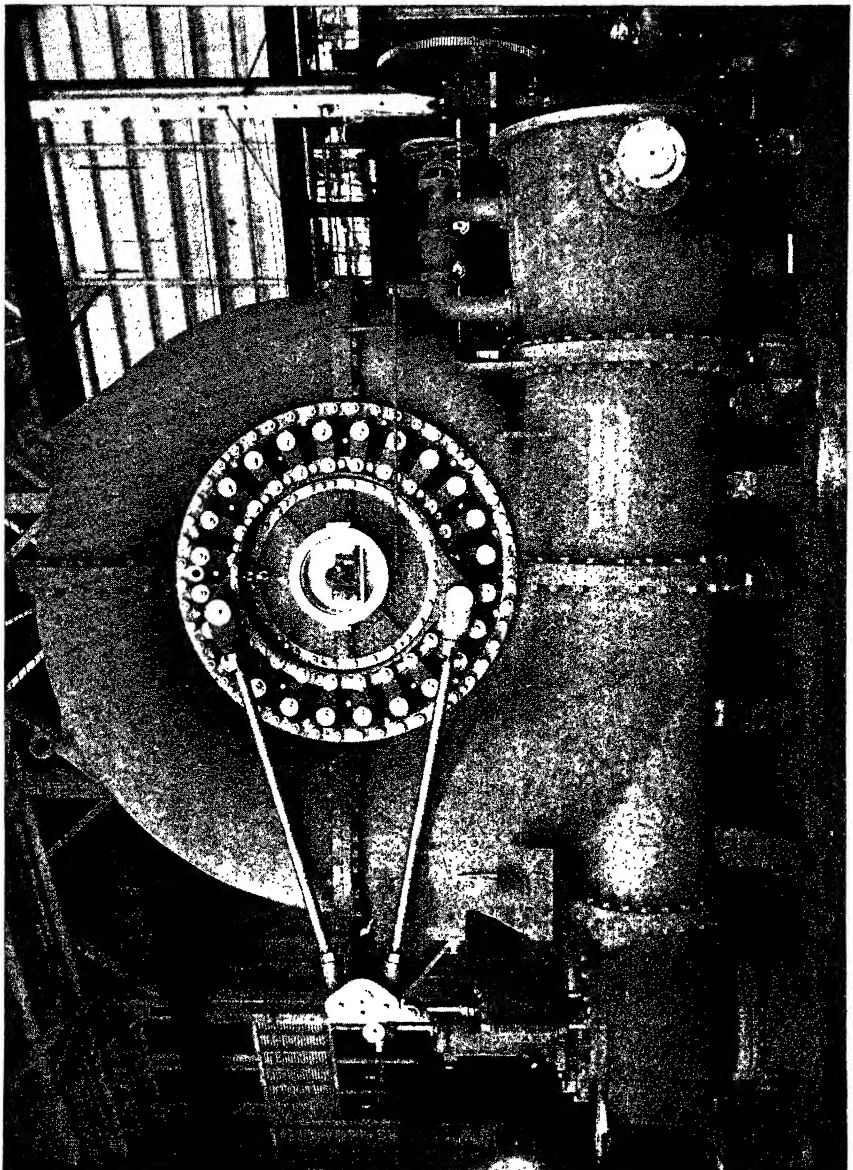


Fig. 115. Single-Discharge Turbine of 20,000 H.P. Operating at 510-Foot Head
Courtesy of Pelton Water Wheel Company, San Francisco, California

sizes, 18-inch to 48-inch wheels, and for timber-frame mounting, or they may be metal covered.

The *Type D* wheels are inexpensive yet serviceable and efficient water wheels for mining and industrial purposes where large initial expense is out of the question. This type is built to operate under heads up to 250 feet, and is supplied in six sizes, 12-inch to 36-inch diameters.

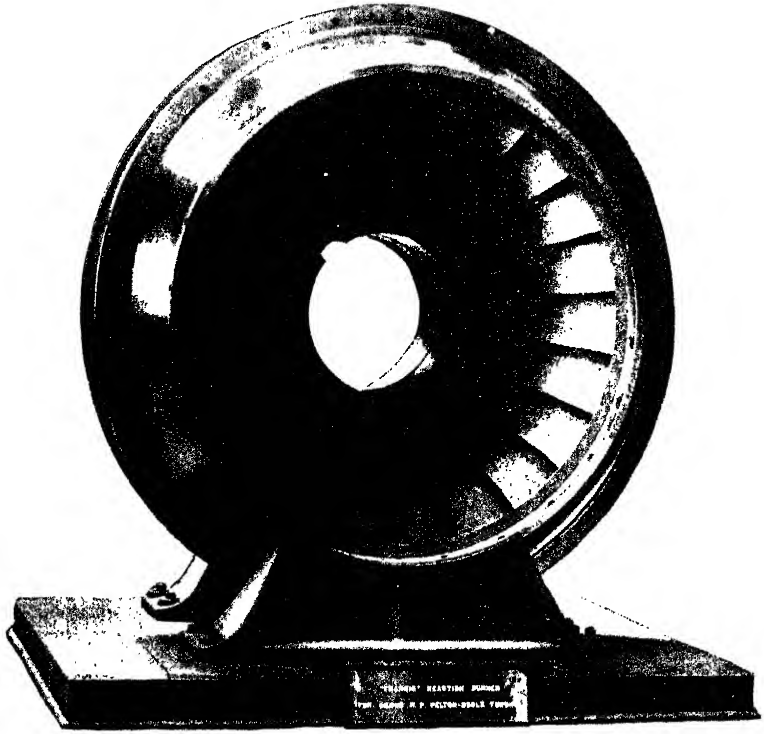


Fig. 116. Single-Discharge Runner for 20,000-H.P. Turbine
Courtesy of Pelton Water Wheel Company, San Francisco, California

Type C motors are metal encased and are suitable for heads up to 230 feet; built in five sizes with 6-inch to 24-inch wheels.

Doble *Ellipsoidal* buckets, Fig. 46, Part I, are of patented form permitting the jet to impinge without shock or disturbance and in connection with which the water is discharged along natural lines over the entire surface. They operate with jets of from ⅜-inch to 10-inch diameter, and under pressure heads as great as 2100 feet.

Classified as *Special* wheels are those direct-connected to air compressors, [Fig. 47, Part I—built in diameters suitable for

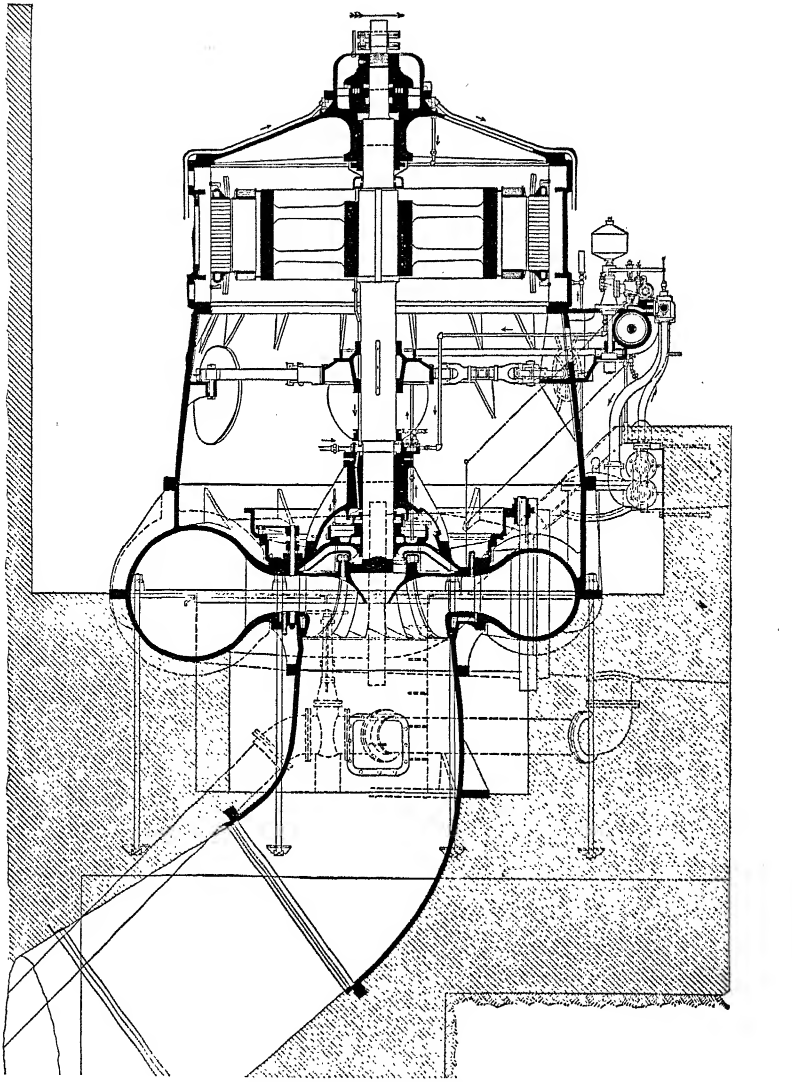


Fig. 117. Sectional Elevation of 500-H.P. Vertical Pelton-Francis Turbine Built for Schenectady Power Company
Courtesy of Pelton Water Wheel Company, San Francisco, California

the operating heads and speeds of compressors. One of the largest sizes installed is a 33-foot wheel developing 1000 horsepower under a

1430-foot head of water. Sheet-steel cover housings are the better practice in such installations.

For electric-current generation the Pelton-Doble tangential water-wheel units, direct-connected, have been designed especially to suit the conditions of the installation and as large as 18,000-horsepower capacity, each, under heads of water up to 1500 or more feet.

Turbines. The Pelton-Francis hydraulic turbines are built in sizes up to 20,000-horsepower capacity in the single-runner single-

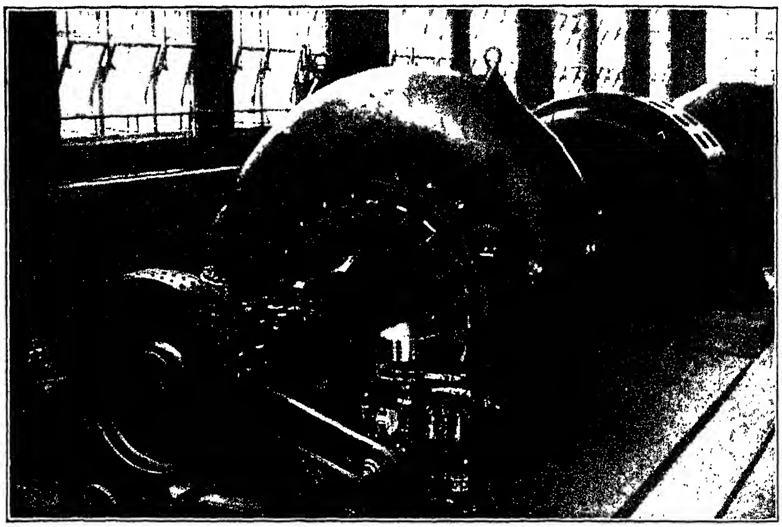


Fig. 118. Two 10,000-H.P. Turbines of the Tennessee Power Company, Operating under 250-Foot Head
Courtesy of I. P. Morris Company, Philadelphia

discharge type, Fig. 115, as well as in other sizes with double runners and other variations.

The spiral-case horizontal-shaft type, Fig. 115, for direct coupling to electric generators has an evolute casing of cast iron or of annealed cast steel; radial inward-flow Francis reaction-type runner—usually bronze—designed for various specific speeds according to conditions, single runner for 200-foot heads and over, Fig. 116, or twin runners for low heads of 50 feet or so; and wicket gates or guide vanes of steel. It may be arranged for single or for double discharge.

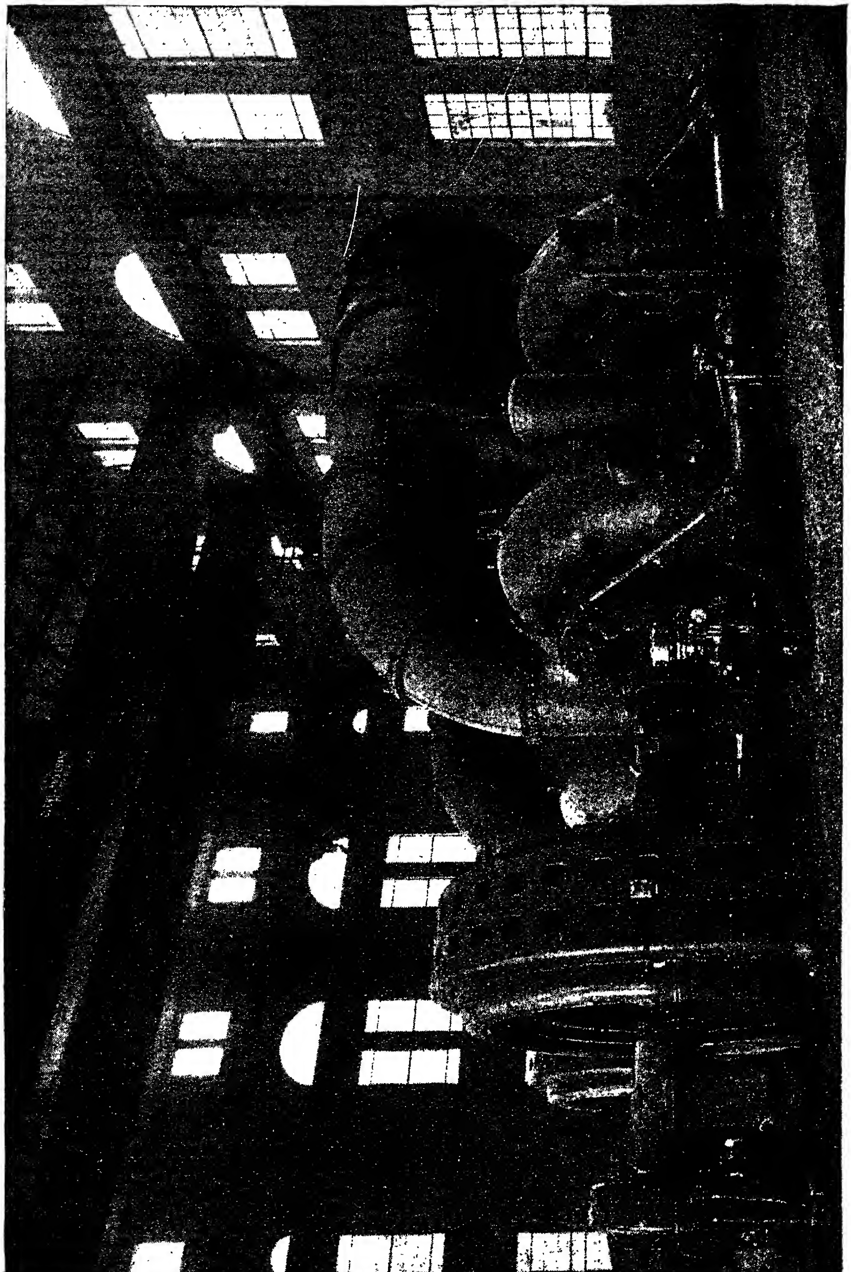


Fig. 119. Two 22,500-H.P. Turbines Built for Washington Water Power Company
These are the highest power turbines ever built. They operate under a head of 168 feet.
Courtesy of I. P. Morris Company, Philadelphia

The vertical-shaft spiral-case type, particularly suitable for modern hydroelectric plants, is supplied with cast volute case and other parts similar to those of the horizontal-shaft types. The weight of the rotating parts is carried by thrust bearings in the top

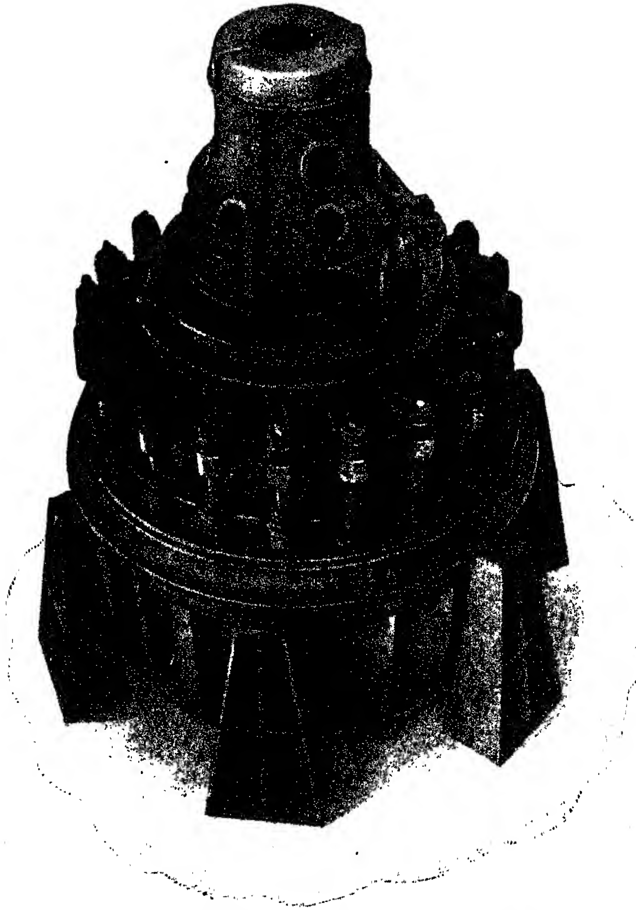


Fig. 120. One of Eight 10,000-H.P. Turbines for Mississippi River Power Company
Courtesy of I. P. Morris Company, Philadelphia

cover plate or yoke. A vertical section of a 5000-horsepower installation of this type is shown in Fig. 117.

108. I. P. Morris Company (operated by the William Cramp & Sons Ship & Engine Building Company). The Morris Company of Philadelphia especially designs its hydraulic machinery in each

case; the method followed being to detail the installation to meet the exact conditions of the particular plant, after careful engineering analysis. The Company's constructions are almost all of the reaction-turbine type. In the selection of the most suitable type of hydraulic machinery, the turbines, as the chief factor affecting the success of a plant, have the following principal points of considera-

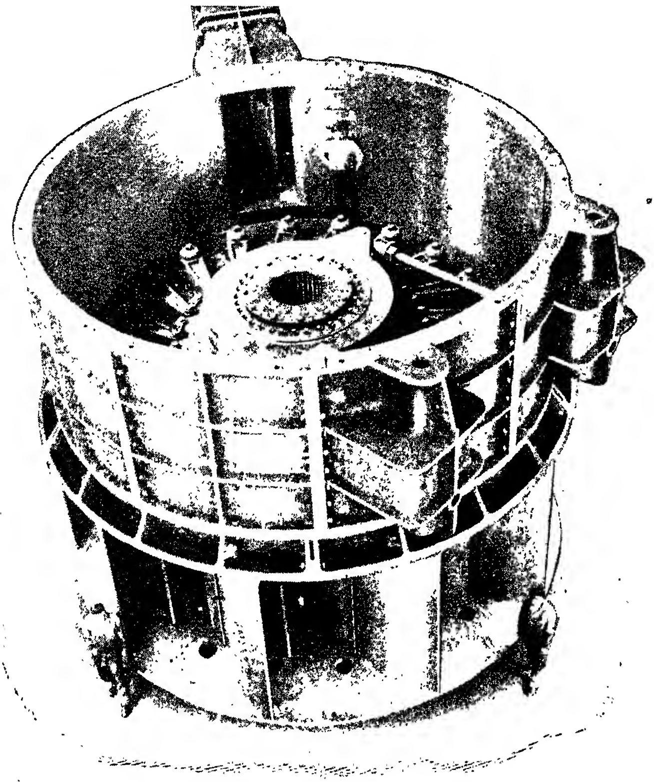


Fig. 121. A 10,800-H.P. Turbine Completely Erected in the Shop
Courtesy of I. P. Morris Company, Philadelphia

tion: *mechanical construction,* requiring great care in design and shop manufacture; *engineering methods in design,* in which proper disposition of weight or strength is the essential; and *operating efficiency,* which is to be realized in practice and not merely stipulated on paper.

Types Constructed. The turbines supplied, operating under many conditions of head from less than 50 feet up, are of various

types: horizontal-shaft single volute-casing type, Fig. 118, with double-discharge runner and two draft tubes per unit; Fig. 119, twin casings, two runners, and a common draft chest and tube; the plate-steel cylindrical end-intake type of casing, with runners discharging into a central common draft chest and tube, conforming to older types of settings; or the modern single-runner vertical-

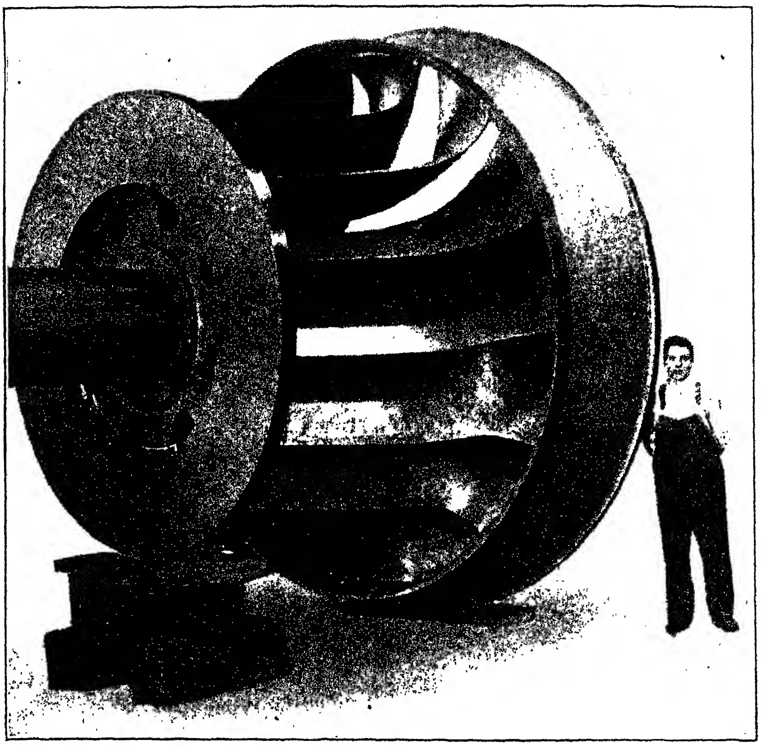


Fig. 122. A 20,000-H.P. Runner for Laurentide Company's Turbines, made of Single Iron Casting
Courtesy of I. P. Morris Company, Philadelphia

shaft type, with volute casings and draft tubes moulded in the concrete substructure.

Among the many installations with features interesting and profitable for study—both on account of physical dimensions and unit capacity, as well as efficiency—are: the massive 10,000-horsepower turbines, Fig. 120 (and see Fig. 212, Part III), of over 90 per cent efficiency, for the Mississippi River Power Company, at Keokuk;

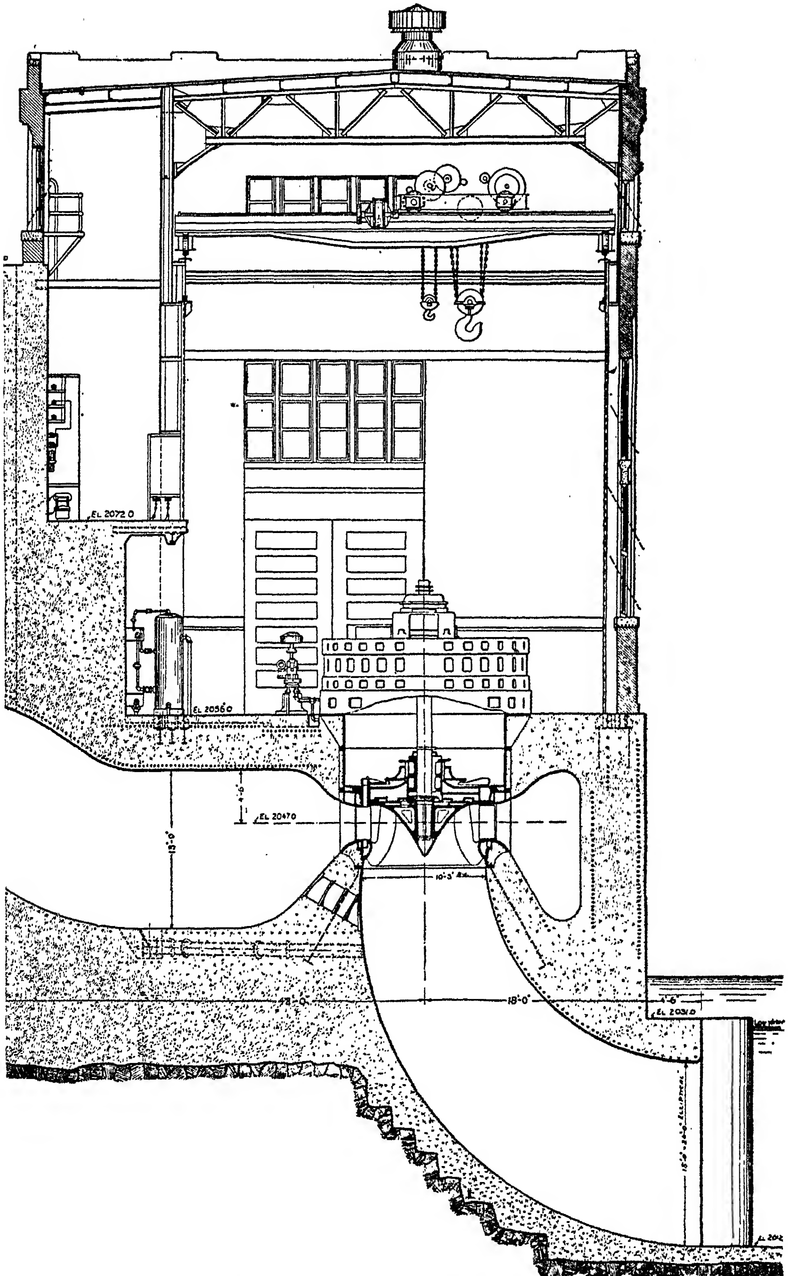


Fig. 123. Sectional View of 6000-H.P. Turbine and Plant of the Appalachian Power Company
Courtesy of I. P. Morris Company, Philadelphia

the 10,800-horsepower units, Figs. 91 and 121—weighing, complete $1\frac{1}{4}$ million pounds each—for the Cedar Rapids Manufacturing & Power Company, on the St. Lawrence River; the 20,000-horsepower vertical-shaft turbines, of the Laurentide Company, Ltd., Grandmère, Province of Quebec, the runner of which is shown in Fig. 122; the 22,500-horsepower horizontal-shaft double-runner turbines, as

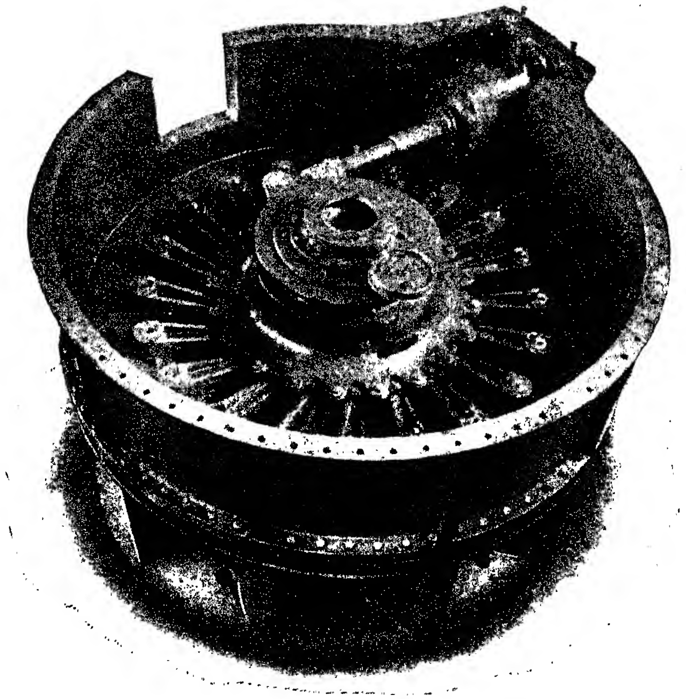


Fig. 124. One of the Turbines for the Appalachian Power Company in Course of Erection
Courtesy of I. P. Morris Company, Philadelphia

shown in Fig. 119, page 169, with an overload capacity of 25,000 horsepower, for the Washington Water Power Company on the Spokane River; the 93 per cent efficiency 16,500-horsepower turbines of the Pennsylvania Water & Power Company, Holtwood, Pennsylvania; and the 93.7 per cent efficiency 6000-horsepower turbines, Figs. 123 and 124, of the Appalachian Power Company, New River, Virginia.

109. Allis-Chalmers Company. The Allis-Chalmers Company, Milwaukee, manufacture several types of turbines to suit the varying conditions found in practice.

Open-Flume Turbines. For the development of water powers having available heads varying in general from 10 feet to 50 feet, where the available site is not restricted or subject to excessive

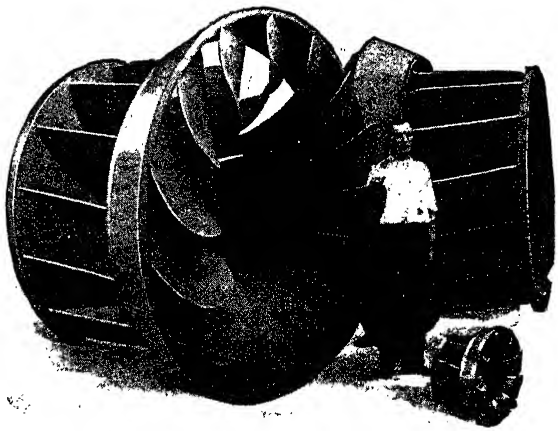


Fig. 125. Typical Allis-Chalmers Turbine Runners

variation of forebay and tailrace level, the open-flume type is applicable, and is manufactured with single, twin, triplex, or quadruplex

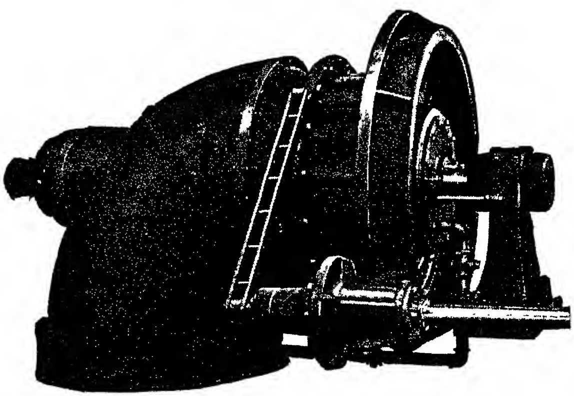


Fig. 126. Single Type "AH" Turbine with Quarter-Turn Draft Tube
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

runners, on horizontal shafts. Figs. 125, 126, and 127 illustrate individual runners and complete machines.

Plate-Steel Case Turbines. In the development of water powers having available heads varying in general from 50 feet to 150 feet,

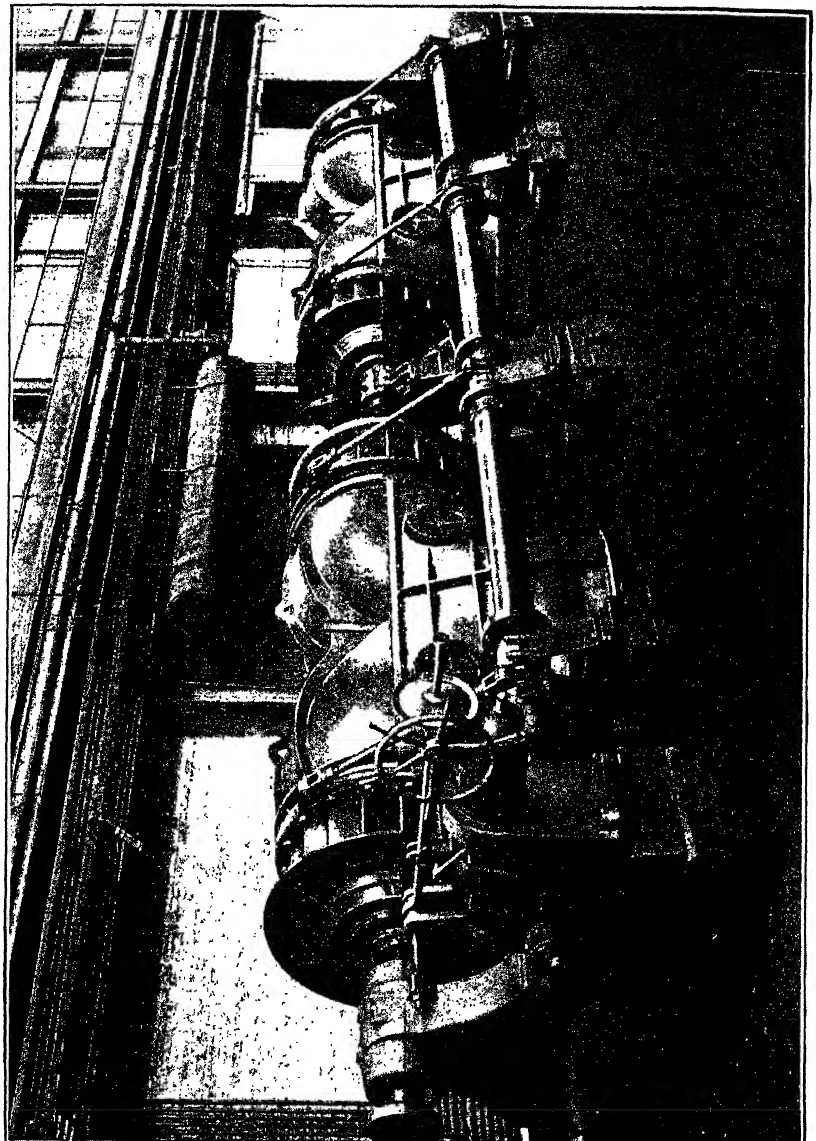


Fig. 127. Quadruplex Horizontal Open-Flume Turbine Built for Crane Falls Power and Irrigation Company
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

the cylindrical case type is supplied as follows: single runner, horizontal shaft, top, bottom, or side inlet; twin runners, horizontal shaft, top or side inlet, Fig. 128, or end inlet; or in vertical settings.

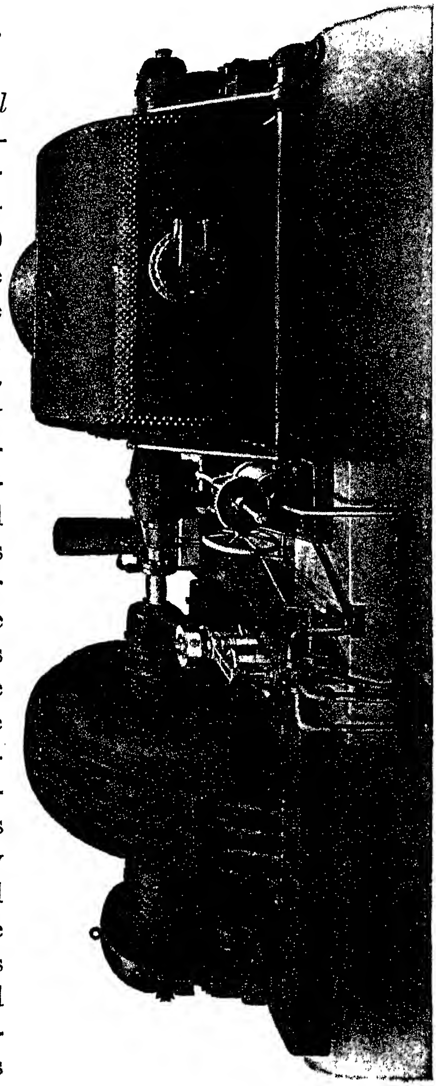


Fig. 128. A 7500-H.P. Turbine for Telluride Power Company, Operating under 125-Foot Head
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

The *spiral plate-steel case type*, Fig. 98, is recommended for the development of water powers having available heads of 50 feet to 150 feet, where the highest economy in the use of the water is required. The design consists of a casing of true evolutionary form built up upon a cast-steel speed ring consisting of two flanges held together by a series of ribs to form a tapered water passage. From its entrance to the casing the water is delivered throughout the whole circumference of the speed ring without disturbance and at uniform velocity. The speed ring serves to increase the velocity gradually to that acquired by the water on its entrance to the guide vanes. This uniform distribution and gradual acceleration eliminates practically all losses due to shock and eddy currents, and enables a plant efficiency to be secured more nearly comparable to Holyoke test-flume results than is possible with any other design. This type is built with single runner or twin

runners and common center charge, horizontal shaft; or single runner in vertical-shaft setting.

Cast-Metal Spiral-Case Turbines. For the development of water powers having available heads ranging in general from 100 feet to 600 feet—the later figure representing at present the approxi-

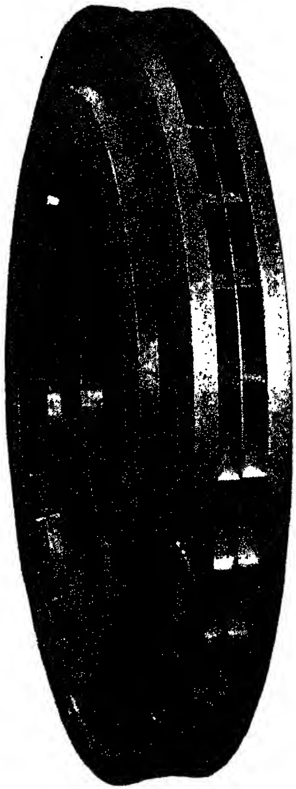


Fig. 129. Typical 8500-H.P. Turbine Runner
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

mate limit of application of the reaction runner, of which a type is shown in Fig. 129—cast-iron or cast-steel spiral-case turbines with the following variations are especially suitable; single vertical-shaft types either with thrust bearing below the generator, or with thrust bearing mounted on the generator cover plate or yoke; single horizontal-shaft type with quarter-turn discharge, or with double

quarter-turn discharge, Fig. 130; and twin horizontal-shaft turbines with each runner in a separate spiral casing and with common center discharge.

MAINTENANCE OF UNIFORM SPEED

110. Problem of Efficient Regulation. In the course of the manifold operations of an industrial plant the quantity of work and the number of machines in actual operation at one time will vary greatly from time to time; nevertheless it is essential that the shafting be maintained at a uniform speed. This is automatically accom-

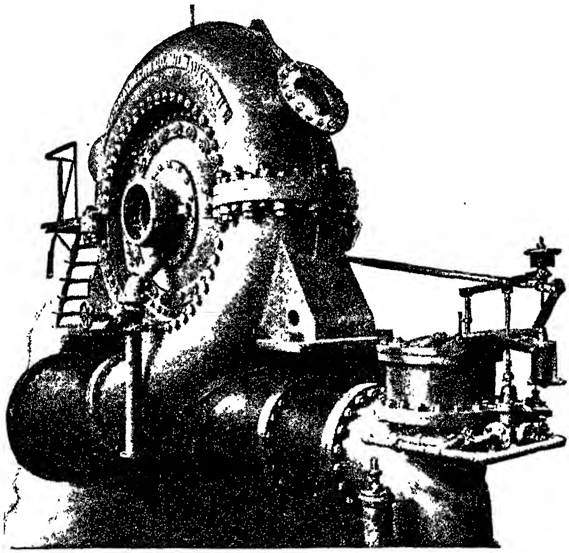


Fig. 130. Allis-Chalmers Turbine with Double Quarter-Turn Discharge and Pressure Regulator

plished by a mechanism or contrivance which regulates the quantity of water passing to the wheels in accordance with the power demand.

One of the most complex problems encountered by the hydraulic engineer is the efficient speed regulation of turbines operating under variable load. Every variation of load or power requires a corresponding quick change in the gate opening, and this involves a change in velocity of flow in the penstock representing a large amount of kinetic energy. A decrease in the gate opening retards the velocity of flow and increases the penstock pressure, and, if this takes place

suddenly, water hammer results and may prove disastrous. Conversely, increasing the gate opening will increase the velocity and reduce the water pressure in the penstock. Thus the variations of gate opening in conformity with variations of load cause pulsations in pressure, or surging of the water.

EQUALIZING DEVICES

For better speed regulation, particularly in the case of long penstocks, special devices for the purpose of equalizing the energy are generally provided.

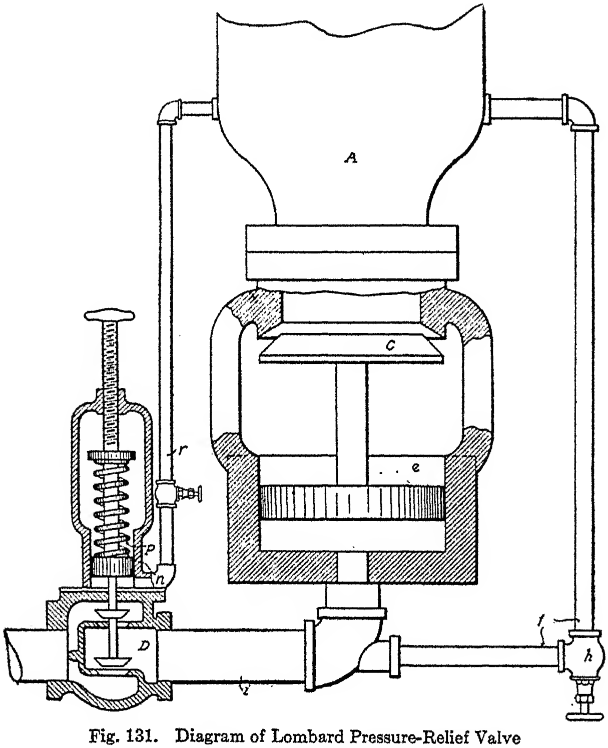


Fig. 131. Diagram of Lombard Pressure-Relief Valve

111. Pressure-Relief Valve. The pressure-relief valve is an auxiliary apparatus which allows water to escape when the pressure rises beyond a set limit. The springs of such a valve require careful consideration in order to permit the valve to open sufficiently under the influence of a relatively small rise of penstock pressure.

The *Lombard* pressure-relief valve is shown diagrammatically in Fig. 131. *A* is the end of the penstock in which the pressure is not to rise above a set limit. The piston *e*, under penstock pressure through pipe *f*, holds the disk of the relief valve *c* to its seat against the water pressure in the penstock, since the piston is somewhat larger in diameter than the disk. The space behind the piston is also connected by means of pipe *i* to the balanced waste valve *D*, held closed by the force of spring *P* against the penstock pressure through pipe *r* behind the piston *n*. The force of the spring *P* is so adjusted that it will be overcome and the waste valve open at any desired penstock pressure. The opening of the waste valve reduces the pressure behind the relief-valve piston *e*, thus opening the relief-valve disk *c* and allowing the water to escape. To perform this function properly, the discharging capacity of pipe *f* must be less than that of pipe *i* and the waste-valve opening, since water begins to flow through pipe *f* as soon as the waste valve opens. A quick closure of the relief valve would result in surging in the penstock, to prevent which the rate of closure is made adjustable by means of the valve *h*.

112. By-Pass. *"The by-pass, which may be employed where economy in water consumption is not demanded, consists of a valve or gate of sufficient area to pass the entire volume of water required by the turbine at full-gate opening, and moved in conjunction with the speed-regulating gates. With the turbine gate fully open, the by-pass is closed; but when the regulating gates commence to close, the by-pass opens, and its passage area is increased in the same proportion as the gate opening decreases, so that the combined area of the gate opening and the by-pass is always sufficient to pass the entire volume of water required by the turbine at full-gate opening; thus the velocity of the water in the penstock and the amount of water discharged remain always the same, the discharge of the by-pass being run to waste. This arrangement not only permits the closest speed regulation with violently fluctuating loads, but also relieves the penstock from shock or water hammer, and is therefore often used in connection with impulse turbines working under high heads and supplied by very long penstocks.

"European engineers have abandoned the ordinary by-pass,

* John Wolf Thurso, "Modern Turbine Practice", D. Van Nostrand Co., New York.

on account of the great waste of water which its use involves, but frequently use the temporary by-pass, which is essentially the same device, except that the speed-regulating gates and the by-pass are connected in such a way that the temporary by-pass will open while the speed-regulating gate closes; but as soon as the closing movement of the regulating gate ceases, the by-pass at once starts automatically to close again slowly, being actuated by a spring, counterweight, or hydrualic pressure. The temporary by-pass does not open at all when the regulating gate closes very slowly. It will

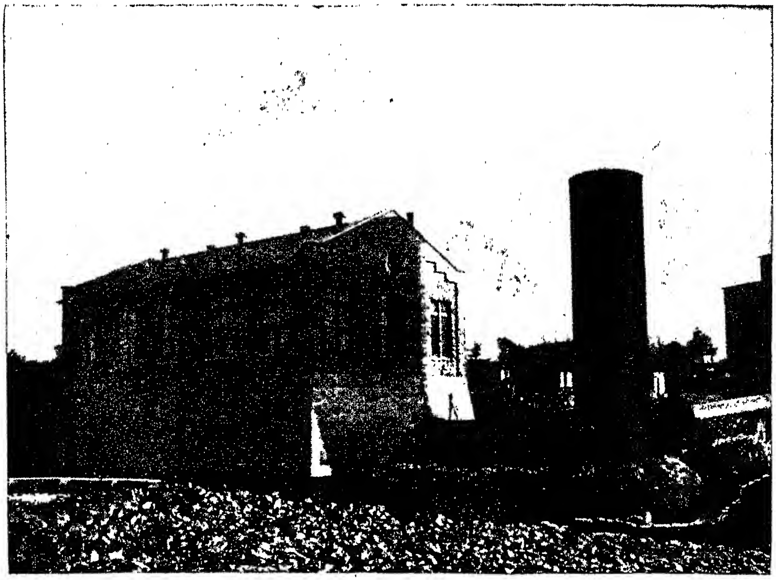


Fig. 132. Power Plant on Big Sioux River with Simple Standpipe
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

be seen that the temporary by-pass is similar in its effect to the relief valve, except that the by-pass opens before a rise in the penstock pressure, due to the closing of the regulating gates, takes place.

113. Standpipe. "The standpipe is frequently employed to aid the governor, and thus to improve the speed regulation of turbines; and is simply an open reservoir which, to a limited extent, will absorb or store energy, when the gate opening is decreased in consequence of a reduction in the load of the turbine, and will supply energy when the gate opening is increased in consequence of an

increase in the load of the turbine. The standpipe is the best possible relief valve, and should have its top edge a few feet above the high-water level in the headrace, and its diameter or capacity should be in accordance with the volume of water discharged by the turbine at full-gate opening, and with the length of the penstock.

"When the gate opening of the turbine is suddenly reduced, the excess of water flows into the standpipe, causing the water therein to rise, and perhaps to escape over the top edge of it, until the water column in the penstock has slowed down; while, when the gate opening is suddenly enlarged, the additional water required is supplied from the standpipe, causing the water therein to fall until the

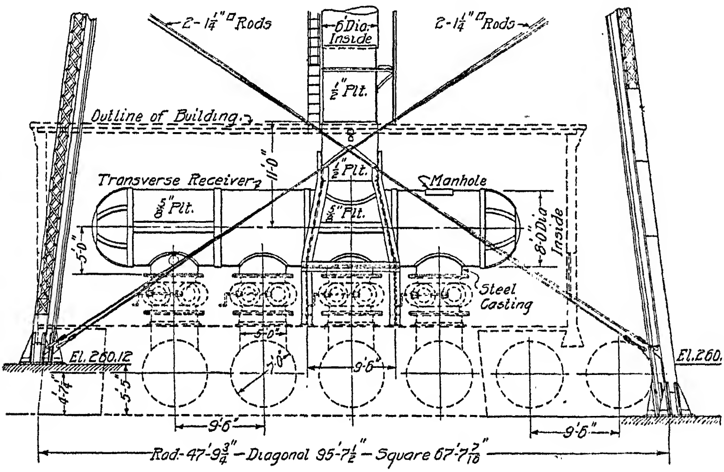


Fig. 133. Details of Base of Surge Tank for Great Northern Power Company

speed of the water column has increased to meet the demand. In connection with high heads, standpipes are rarely used, as they are not only very expensive but also less effective on account of the inertia of the water column in the standpipe.

"Standpipes must be carefully protected from freezing, as otherwise they may be rendered entirely useless. A waste pipe should be provided to carry off the water escaping over the top edge of the standpipe."

In the installation of the Sioux Falls Light and Power Company, at Sioux Falls, South Dakota, the water is carried from the forebay to the power house through 680 feet of 7-foot steel penstock.

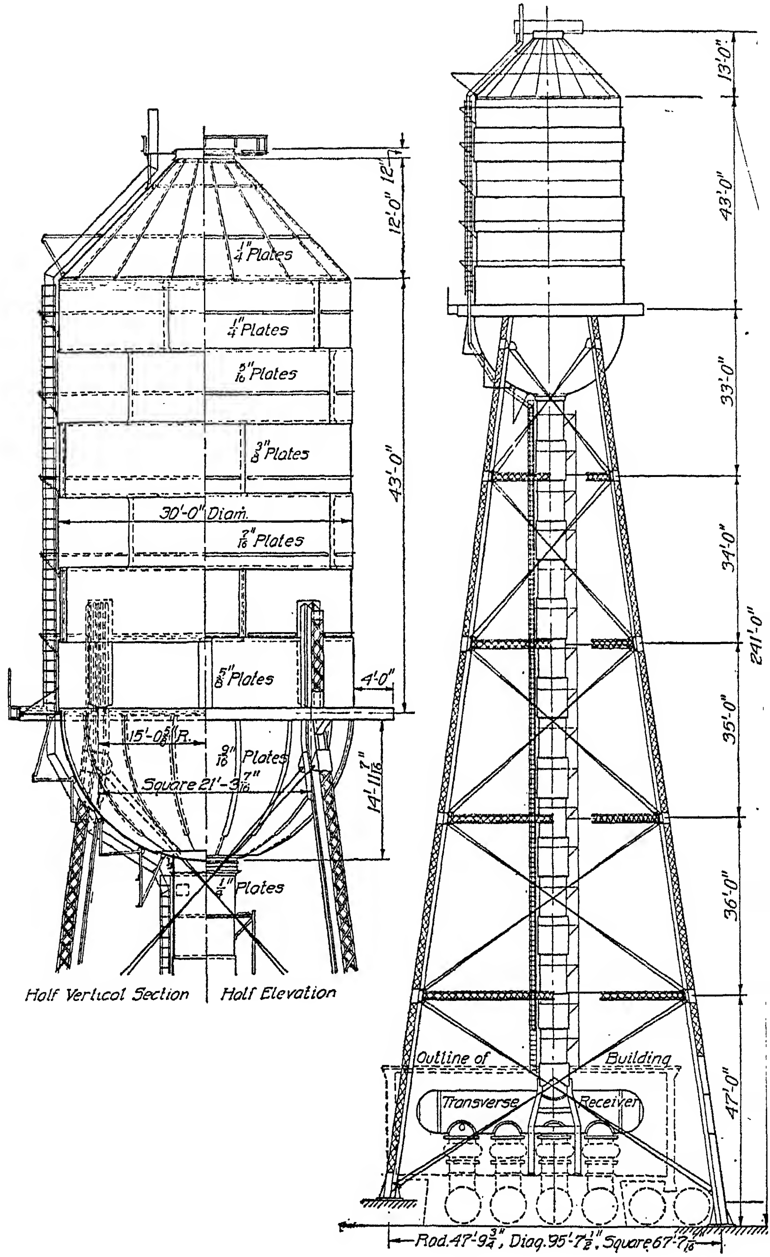


Fig. 134. Details of Surge Tank and Tower for Great Northern Power Company

At the foot of the penstock there is a plate-steel standpipe, as shown in Fig. 132, 16 feet in diameter and 62 feet high, of sufficient capacity to supply water for sudden fluctuations in the load without excessive pressure variation in the pipe line. During the operation of the plant, heavy fluctuations on the street-railway load, resulting from snow blockades, have caused a maximum difference of water level in the standpipe of only 18 inches.

114. Surge Tank. In some cases surge tanks are employed in which the movement of the water is throttled down by the interposition of resistance to the flow. In the hydroelectric development of the Great Northern Power Company of Duluth, the surge tank is located about 500 feet from the power house at the top of the bluff. At this point each pipe line is connected by a header and valve with a cross receiver, Fig. 133. This in turn is connected to a 6-foot standpipe 235 feet in height from the bottom of its foundations to the top, Fig. 134, where it is enlarged to form a steel tank 30 feet in diameter and 70 feet high. The function of this standpipe is to relieve increase of pressure caused by closing of the turbine gates and also to prevent pressure drop on the sudden opening of the gates. The water in the standpipe is kept from freezing by the circulation of hot air in the space between the pipe and the lagging, the heat being supplied by steam from a heating plant located at the foot of the pipe.

115. Air Chamber. Sometimes air chambers are used on penstocks for protection against water hammer, but they are not efficient in turbine regulation. Air chambers should be of ample capacity, and should be connected with an air pump to replenish the air; try cocks and gage glasses should be provided for observing the working of the chambers.

The safety devices described above should be placed at the lower end of the penstock where water hammer is first manifested and has its greatest effect.

116. Flywheel. Occasionally flywheels are employed to assist the governor in speed regulation of turbines operating under high heads, and in this capacity are useful in eliminating small momentary speed variation due to various causes which the governor cannot control. Sometimes the armature or revolving field of a turbine-driven dynamo may be made to serve as a flywheel.

GOVERNORS

117. Mechanical and Hydraulic Control. "The means for controlling the speed of a hydraulic turbine under the various changes of load all have for their ultimate result the control of the quantity of water operating the turbine, whether this is of the impulse or of the reaction type. The power to operate the turbine gates is sometimes furnished by the turbine itself; but on account of the size and weight of these gates in a large installation, considera-

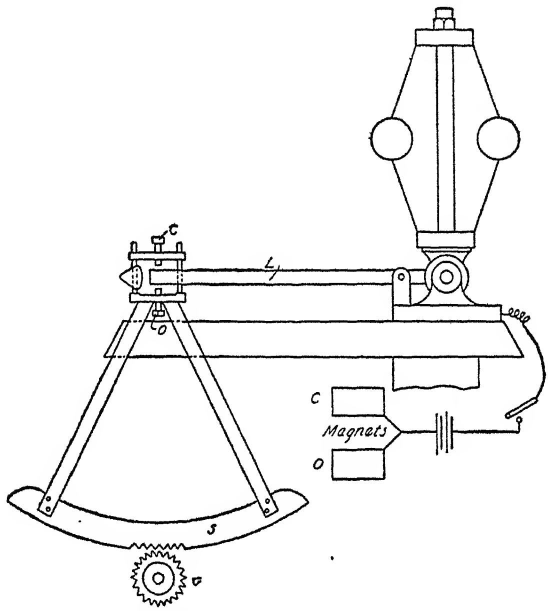


Fig. 135. Diagrammatic Representation of "Replogle" Centrifugal Governor and Return

ble force must be exerted to operate them; and for this reason, they are not directly actuated by the sensitive centrifugal governor. An auxiliary relay motor, using either mechanical or hydraulic power, which is itself controlled by the centrifugal governor is therefore interposed for the purpose of actuating the gates."

Relay Motor. "The hydraulic relay consists essentially of a piston and cylinder operated by pressure water from the penstock, or oil from a pressure tank. The governor proper consists of the usual type of revolving centrifugal balls in gear with the turbine shaft, and so connected with the valves or devices of the motor or

other mechanism which actuates the gates, that any change in the relative position of the balls brought about by a variation in the turbine speed, will bring into play the mechanism which operates the gates." (See also Article 64, Part I, page 8.)

Over Governing or Racing. When a change of speed due to a change in load takes place, the governor will set the regulating gate in motion; but, owing to the inertia of the water, a certain amount of time is required for the turbine to return to the proper speed; so

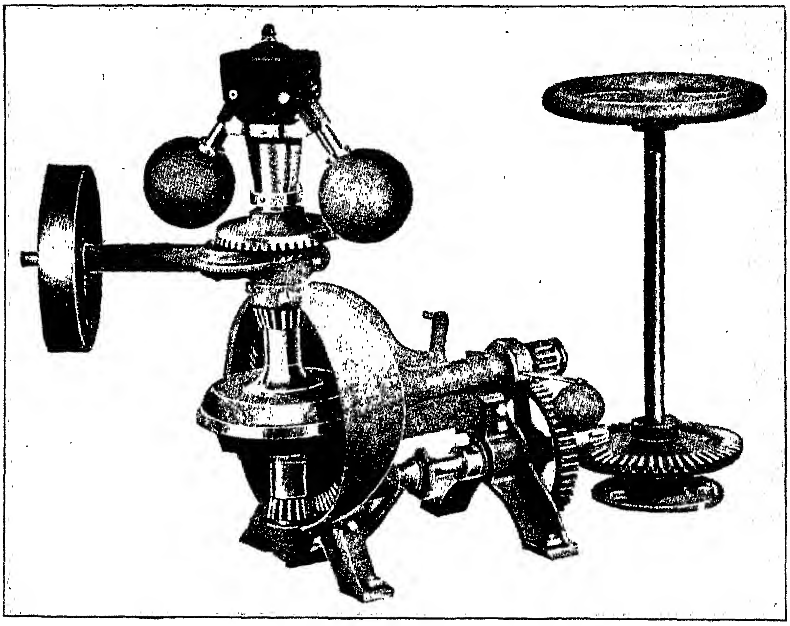


Fig. 136. Standard Type Woodward Governor Suitable for General Mill Work

that the gate would have traveled in the interval beyond the required position, and in a short time, under the influence of the governor, would start moving in the opposite direction. To obviate this difficulty, the motion of the relay, and with it the movement of the regulating gate, must be arrested before the latter has traveled too far, by a return device.

Time of Closure. Mechanical governors effect an entire closure of gate usually in 15 to 25 seconds, although some types have been constructed that require only about 3 seconds.

In the case of hydraulic governors, this time may be reduced to one second or a fraction of a second; such governors therefore afford very close speed regulation.

118. Replogle Centrifugal Governor and Return. This device represented diagrammatically in Fig. 135 is well known in American practice. S is a toothed segment operated by the turbine-gate

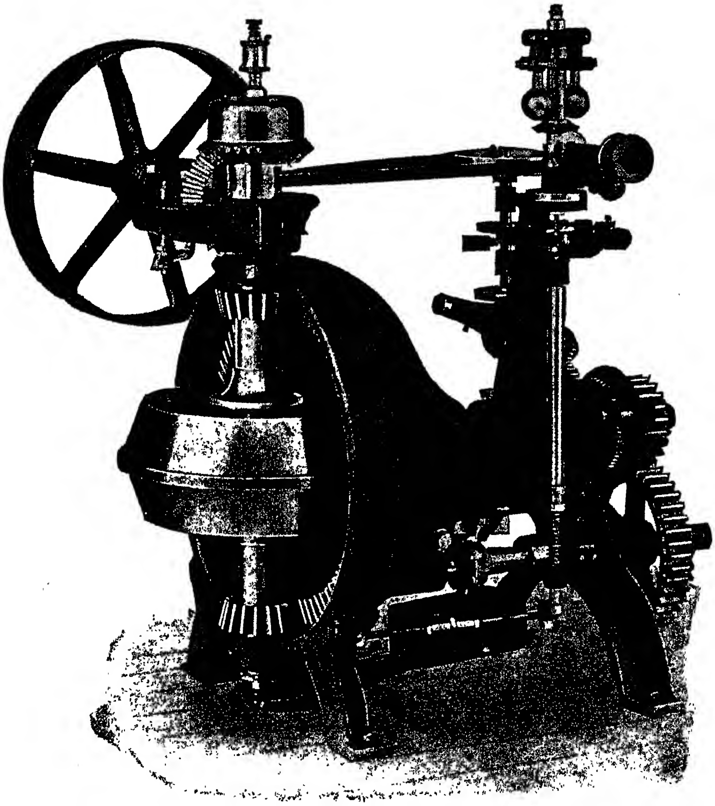


Fig. 137. Compensating Type Woodward Governor Suitable for Electric and Heavy Mill Work

shaft G, which is also toothed for the purpose. L is a lever, tilted up or down by the governor balls when a variation of speed occurs; this lever, in its motion, completes an electrical contact by touching points C or O. Such contact energizes one of two electromagnets in the circuit, by which means an auxiliary machine, or relay (not shown) is tripped, and begins to open or close the turbine gate by

turning G in the proper direction. The turning of G carries the rack S with it, which, by a suitable cam arrangement, breaks the contact between the lever and the point C or O. This interruption

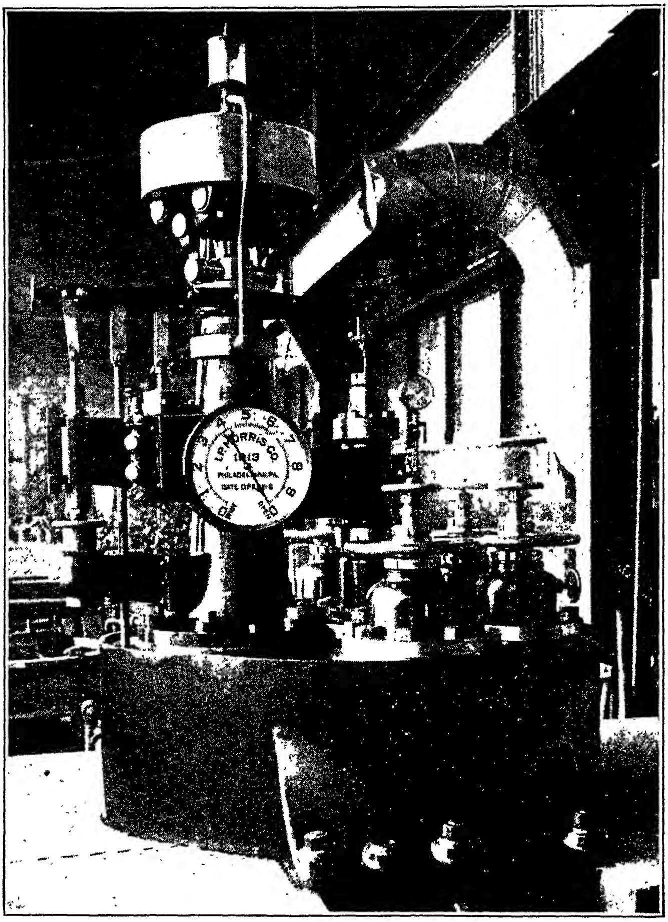


Fig. 138. One of Twelve Governors for Cedar Rapids Manufacturing and Power Company
Courtesy of I. P. Morris Company, Philadelphia

of the circuit cuts the auxiliary power out of action, thus stopping the motion of the gate before the increase or decrease of power due to the gate motion has made the turbine speed above or below normal. Later types of this governor have been still further refined.

119. Woodward Governor. Figs. 136 and 137 illustrate two types of Woodward governor.

120. Glocker-White Governor. This governor possesses a novel feature. There are two chambers connected by a small opening in the hollow governor balls which are partly filled with mercury. At normal speed the axis of the balls is vertical, and the mercury is equally divided between the two chambers. With increasing speed the balls move away from the axis, and some mercury flows from one chamber into the other by reason of the centrifugal force. The center of gravity being thus raised, its lever arm and consequently its effectiveness is increased, since its movement increases in more rapid ratio than the speed; and the reverse condition is also true.

121. I. P. Morris Company's Double Floating-Lever Type. The governor now furnished with this company's turbines was developed for the purpose of securing a machine which would harmonize with the design and construction of its turbine wheels. The illustration, Fig. 138, depicts one of twelve governors built for the Cedar Rapids Manufacturing & Power Company, Canada, for the control of vertical-shaft turbines each having a capacity of 10,800 horsepower when operating under a head of 30 feet.

The fly-ball mechanism is of very heavy construction. The governor proper is mounted on a valve chest in which are located the four control valves which connect the governor to the pressure system and to the operating cylinders or servo-motors of the turbine.

The secondary governor valve is also located within this valve chest. The primary valve which operates the secondary valve is shown to the right of the gate opening indicator. The stem of the pilot valve which controls the primary valve may be seen connected to the right-hand end of the long horizontal floating lever. The name of the governor, "double floating lever", is derived from the fact that there are two floating levers: the long horizontal lever, and the small horizontal lever, which can be seen within the larger one on the left. The former is attached at its center to the governor collar, which is in turn connected to the fly balls in the governor head.

The small floating lever has three connections, one to the dash pot on its right-hand end, one to the gate mechanism of the turbine at its center, and the third to the compensating device on its left-hand end.

The four valves in the valve chest are of special design and are equipped with by-passes. The small hand wheel at the top, above the main hand wheel of each valve, controls the by-pass. This construction was found desirable because, if ordinary 7-inch gate valves were used, one man could with difficulty operate under a working pressure of 200 pounds.

122. Allis-Chalmers Hydraulic Type. The Allis-Chalmers Company, after extended investigation, decided to develop the

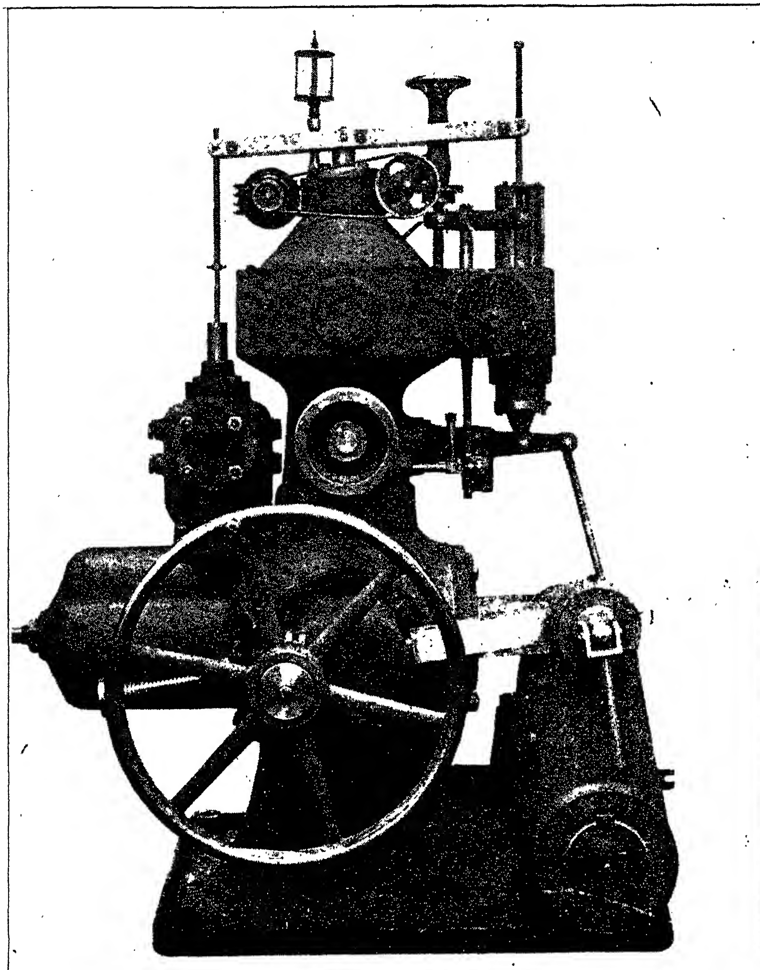


Fig. 139. Standard Horizontal Hydraulic Governor
Courtesy of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin

hydraulic type of governor exclusively, Fig. 139. Their description in part follows:

For supplying the energy, a rotary pump is used to force oil under pressure into a welded-plate pressure steel storage tank the upper part of which contains

air under pressure. The rotary pump consists of a pair of smooth running gears, one driven from an external source, and this in turn drives the other. The gears are so well fitted that air can be readily pumped along with the oil, making the use of a separate air compressor unnecessary except in connection with large central oil systems, where the air capacity is very large. For small units where not much oil is required a safety relief valve is mounted on the tank and can be adjusted at will to suit any pressure, and it discharges back into the receiving tank that amount of oil which is not required by the governor.

Where a large amount of oil is required a central oil system is provided having duplicate pumps which force oil into the pressure tanks located near each governor. The large size pumps are driven directly by motors. In order to reduce the power required on this larger-size pump an *unloading valve* is provided, which by-passes the oil directly back into the receiving tank while the pressure on the pump is reduced to practically nothing.

For distributing the energy the governor proper is used, consisting of fly balls, regulating valve, and relay compensating device. The fly balls, which are mounted in a dust- and oil-proof casing and are self-lubricating, either raise or lower the floating lever in accordance with the speed change. The motion of this floating lever is communicated to the *regulating valve,* which is hydraulically balanced, and consists of a main piston valve and a pilot valve, the latter covering small ports in the former. A slight movement of the pilot valve relative to the main valve distributes the oil on either the upper or lower end of the latter, which in turn moves to a position where the valve rod again covers the small ports. The result of this action is that the main valve is moved practically the same amount as the pilot valve, although the energy required is just sufficient to move the latter. The pressure from the storage tanks is led to the regulating valve where it is distributed either to the opening or closing side of the regulating cylinders.

In order to provide for stopping the action of the gate mechanism before the speed has returned to normal under the actions detailed above, there is added a compensating relay dashpot, which serves to bring the regulating valve back into mid-position as soon as the gates have started to move in either direction. For large load changes the valve is deflected considerably, which necessitates the dashpot moving through a greater range in order to bring the regulating valve back into mid-position. For this reason a spring and small valves arranged in the dashpot piston serve to bring this dashpot back to its mid-position as soon as possible, depending upon the fly-ball effect of the connected load and the pipe-line conditions. Two relays are provided to complete the control.

For applying the energy to the gate mechanism for small units, regulating levers are placed on the generator floor and connected through piston rod and connecting links to a regulating shaft which transmits the energy to the guide vanes themselves. For large units the regulating cylinders are placed directly on the turbine, and pipes lead the oil from the regulating valve to the regulating cylinder.

123. Pelton Governing Devices. The Pelton governors are of two distinct types:

(1) *Direct-Motion.* This type of governor imparts its energy to the wicket gates of the turbine or the nozzle mechanism of the

impulse wheel without intervening levers. This type of governor is invariably furnished with separate pumping unit and accumulator tank containing oil under air pressure.

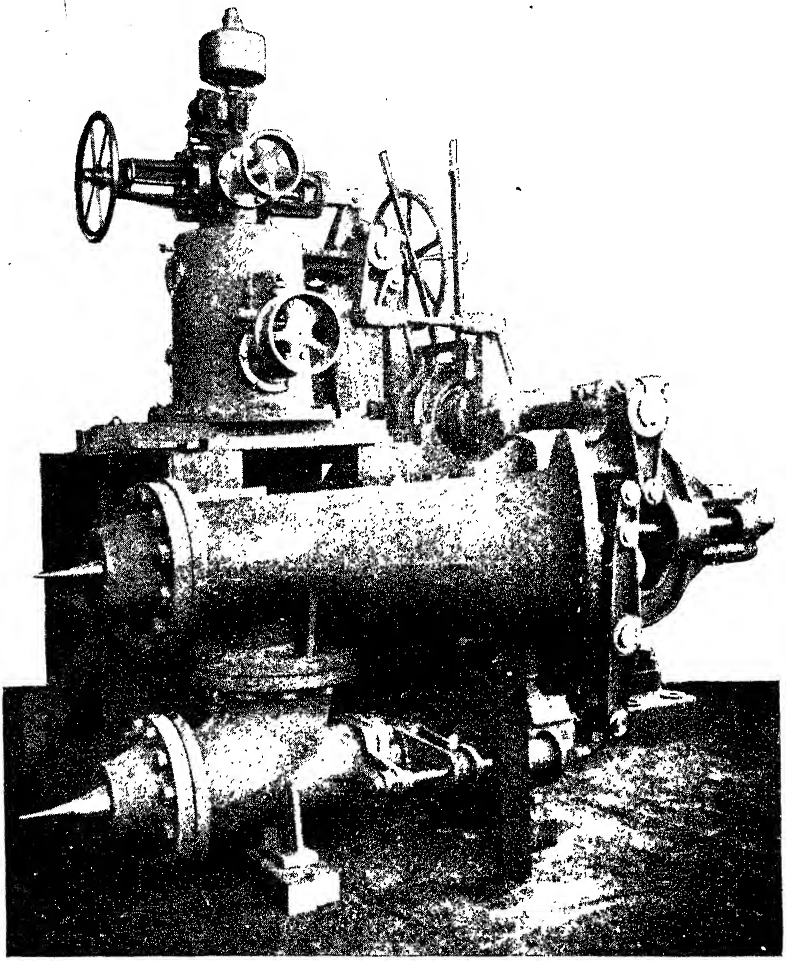


Fig. 140. Pelton-Doble Governor with Automatic Relief Nozzle Attachment, the Latter Being Designed to Bring About a High Economy of Water in Governing
Courtesy of Pelton Water Wheel Company, San Francisco, California

(2) *Reciprocating.* A reciprocating type of governor together with an automatic relief nozzle attached to it are shown in Fig. 140. This combination is the highest type of water economizing device, and where properly installed and adjusted, only a very small per-

centage of the water is wasted in governing. Its operation is as follows: Only the jet from the upper nozzle strikes the buckets of

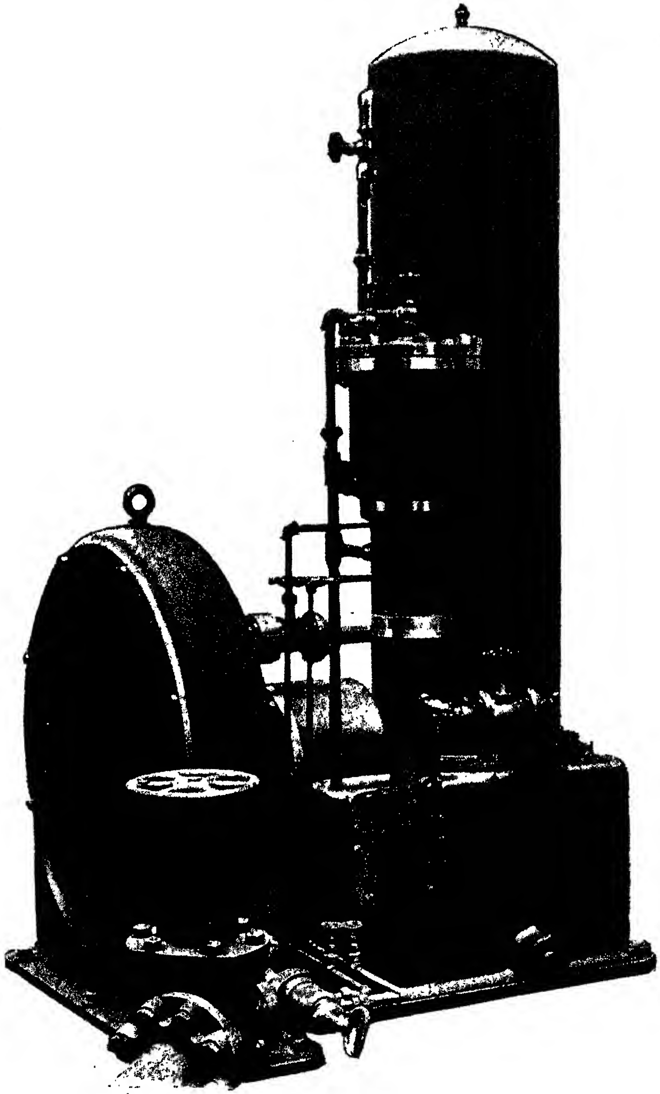


Fig. 141. Independent Accumulator and Motor-Driven Pump for Governor Oil-Pressure System
Courtesy of Pelton Water Wheel Company, San Francisco, California

the wheel, the jet from the lower nozzle being directed against a baffle plate, usually of the Ensign type, placed on the opposite side

of the tailrace. Assuming that the sudden reduction of load occurs, the governor limit will close the upper needle. Should the rate of closing, in order to take care of this sudden fluctuation of load, be of such magnitude that the resulting surge in the pipe line would be excessive or approach the point of rupture, the lower needle will be opened to discharge exactly the same amount of water as is cut out from the upper nozzle. The needle of the lower nozzle is then slowly closed by means of coil springs of the compression type, this closing being accomplished at a rate just sufficient to prevent excessive pressure rises.

Referring to Fig. 140, the governor connection and bell crank lever to the needle nozzle are plainly indicated, while the link motion to the lower needle is also very plain. The link, however, does not attach direct to the lower needle, but instead attaches to the piston rod of a piston operating in an oil cylinder. The two ends of this cylinder are connected through small needle valves which adjust the rate of flow from one end to the other. Thus if the motion of the link and piston is slow, the piston will merely force the oil from one end of the cylinder to the other without operating the needle. Should the motion be quick, however, and the oil be unable to escape through the small ports available for its passage, the piston will be held in its position by the oil in the cylinder, and the coil spring opposing the motion of the lower needle will be compressed and the needle opened.

The independent accumulator and pump for the governor oil-pressure system is shown in Fig. 141. This consists of a small water motor direct connected to a positive displacement rotary pump which maintains the oil at a constant level, as well as at a constant pressure, in the oil reservoir. Floats operating in the chambers, shown in the fore of the reservoir, control the admission of oil under pressure to either side of a cylinder surrounding the stem of the needle valve which is shown in the foreground. The water motor is operated by a hand valve.

When the oil drops to a certain level, the needle is automatically opened, thus starting the water wheel. When the oil has reached a certain level in the storage tank, this needle is automatically closed. Suitable air connections are made, that an adequate supply of air may be maintained in the reservoir.

PRESENT PRACTICE IN DESIGN AND CONSTRUCTION*

124. Features of Improvement. During the past ten years, hydraulic-turbine design has passed through a stage of wonderful development and, in respect to many features, remarkable progress has been made toward bringing the turbine to a high state of perfection, e.g.:

(1) As a result of theoretical investigation, substantiated by experimental research of the highest order, efficiencies have been increased on the average about 10 per cent.

(2) Corrosion of runner vanes, at one time a serious factor as affecting the maintenance of turbines and their efficiencies, has been eliminated subsequent to conclusive demonstration that it is primarily the result of defective design.

(3) The development of high specific-speed runners, thereby permitting the use of higher rotational speeds, has resulted in the rapid passing away during the past few years of the multirunner types of turbines in favor of the vertical-shaft single-runner type.

(4) At one time it was generally thought that the character of speed regulation of a turbine was dependent entirely upon the governor. Today, however, it is known that the make or type of governor has little or nothing to do with speed regulation, except in so far as it must have sufficient power and proper mechanical adjustments to move the turbine gates in a certain time and without hunting.

EFFICIENCY

125. Comparative Increase. Ten years ago it was considered a notable achievement to obtain in a turbine in place, an efficiency as high as 82 per cent. The average of the best results secured was in the neighborhood of 80 per cent. The maximum guarantees of the builders were from 78 per cent to 80 per cent, and were considered highly satisfactory.

During the past two years, efficiencies between 89 per cent and 92 per cent have been quite common, while a maximum value of 93.7 per cent has been secured. Today the builder must guarantee an efficiency of at least 87 per cent in order to secure business.

* From a paper read before the Canadian Society of Civil Engineers, January 15, 1914, by H. B. Taylor, Member, American Society of Mechanical Engineers.

Factors. This remarkable increase in efficiency is not due entirely to superior runner design. As a matter of fact, the improvements in the design of wheel casings, wicket gates, draft chests, and draft tubes have increased the efficiency of the turbine as much as the more efficient runners.

In comparing designs prepared 10 years ago with recent designs, the most striking feature is the absence in the latter of sudden turns and abrupt changes of section in all water passages from the turbine casing to the tailrace; and although in the older designs these passages were thought to be consistent with good results, they now appear to be exceedingly crude. The elimination of losses due to eddies and whirls, abrupt changes of section, and sudden curvature in the various parts which constitute the turbine has improved the efficiency of the turbine as a whole.

126. Influence of Tests. By means of the Pitot tube, it has been possible to determine accurately the conditions of flow in the various parts of the turbine casing, through the guide vanes, in the space between the guide vanes and the runner, and also in various sections of the draft tube. Thus it has been possible to compare actual conditions of flow in these various parts with the calculations.

The water-wheel builders in the past five years have carried on very extensive tests at the Holyoke testing flume in Massachusetts on small runners ranging, in most cases, from 12 inches to 36 inches in diameter. These runners are exact models of the large runners which the builders contemplated installing in the turbines. Various designs of wicket gates and draft tubes are used in connection with the testing of these models. The results of these tests have given the builders very valuable data upon which to base the design of any new work calling for runners of like characteristics. The builders have also been able to secure from these tests empirical constants by which they have been enabled to modify their theoretical data.

127. Complete Plant. In many of the power developments carried out a number of years ago, and in some few instances in more modern plants, even such efficiency as could be obtained from the turbines cannot be realized in the operation of the plant as a whole. Poor designs of the waterways leading to the turbine casing and receiving the discharge from the draft tube cause considerable losses

outside of the turbine passages. In some instances, much attention has been given to slight variations in turbine efficiency, while losses of head amounting sometimes to 1 or 2 feet, occurring through submerged arches, bridge piers, or racks, have been overlooked.

Many recent developments are being much more carefully worked out, so that there are gradual transitions in water velocity at entrance to the turbine casing and at discharge from the draft tube into the tailrace; and practically the entire efficiency of the turbine is realized in the operation of the complete plant.

DETERIORATION

128. Mechanical Reliability Important. Accompanying a growing appreciation of the value of efficiency, engineers are giving increased attention to the mechanical features of turbine designs and to questions of deterioration of the machinery during operation. It is now realized, especially by those who have been taught in the school of bitter experience, that mechanical reliability and stability of construction are of even more importance than efficiency, since continuity of operation and freedom from interruption for repairs are vital considerations. Among other sources of interruption to service, the deterioration of runner vanes has in the past been a frequent cause of trouble.

129. Corrosion of Runner Vanes. Corrosion or pitting of runner vanes, guide vanes, and draft tubes, is no longer a serious obstacle to be contended with in the operation of hydraulic turbines, as its cause is now pretty well understood and can be eliminated by correct designing. Turbine builders do not hesitate to design and construct turbines for operation under any head up to 650 feet and to guarantee their lasting qualities, providing the speed and power in each case are so chosen as to be consistent with favorable hydraulic conditions.

Corrosion or pitting must be distinguished from erosion, which is mechanical wear due to the presence of foreign substance in the water. Erosion is entirely a mechanical action, while corrosion is the result of chemical action. The abrasive action of foreign substances in the water has the effect of first polishing the vane surfaces, and eventually of cutting away the metal until the vanes are worn entirely through. The eroded parts are, therefore, smooth, and can

be readily distinguished from the pitted marks which result from corrosion.

HIGH-CAPACITY RUNNER TYPE

130. Change of Type. Until 1912 a very large majority of turbines applied to low heads were either of the vertical-shaft multirunner type, or of the horizontal-shaft multirunner type. The reason for using more than one runner on the shaft, which is not desirable from a mechanical standpoint, was that this arrangement permitted high rotational speeds and, therefore, economic sizes of generators.

During the past two years, however, the general adoption of the single-runner vertical-shaft turbine for low and medium heads has been witnessed. This change in the type of unit has been made possible by recent progress in the design and development of high-capacity runners. Thus, for a given head and speed it is now possible to secure from a runner a greater output than was possible 2 or 3 years ago; or, conversely, for a given head and capacity it is possible to operate the more recently designed runners at a much higher rotational speed than was the case with runners designed a few years ago. This increase in capacity of runners has been secured without sacrifice of maximum efficiency and with only a small sacrifice in efficiency at fractional loads.

131. Specific Speed. In referring to the capacity of a runner, there is a certain relation between head, speed, and power output which is best expressed by a theoretical characteristic known as "specific speed" N_s.

The *specific speed* of a runner may be defined as *the speed at which any runner will operate when it is reduced to such a size that it will develop 1 horsepower when operating under a head of unity.* The numerical value, expressed in the metric system, of the specific speed of any runner may be found by first calculating the speed and power output of the runner under consideration for 1 meter head, and then mathematically reducing the runner in size until it will deliver 1 horsepower. The speed of this reduced runner, when operating at its point of maximum efficiency, is its rated specific speed, and this speed is expressed in the formula

$$N_s = \text{r.p.m.} \times \frac{\sqrt{\text{HP.}}}{h^{\frac{5}{4}}}$$

the quantities being expressed in the metric system. For converting this to the foot-pound system of units, the formula is

$$N_s = 4.45 \times \text{r.p.m.} \frac{\sqrt{\text{HP.}}}{h^{\frac{5}{4}}}$$

in which the horsepower and head are expressed in the foot-pound system, while the N_s remains in the metric system. The specific speed as calculated from this latter formula is the one generally used. In figuring the specific speed of any turbine having more than one runner, the horsepower used in the formula, of course, is the output of each runner.

An examination of the formula will show that for a given r.p.m. and head, the hp. output is proportional to the square of the specific speed; also that for a given head and hp., the r.p.m. of a turbine is proportional to the specific speed.

Five years ago a specific speed of 275 was considered to be quite high, while today a specific speed of 400 is secured together with as high a maximum efficiency as was previously secured with a specific speed of 275. Thus, today, it is possible with very high efficiency to secure for a given condition of head and power, and with the same maximum efficiency, a speed in the neighborhood of 50 per cent greater than was possible 5 years ago; or for a given head and speed, an increase of approximately 100 per cent in the horsepower can be obtained.

With the highest specific speed which could be secured 5 years ago, the resulting r.p.m. for a vertical-shaft single-runner turbine for low heads was, in a great majority of cases, such as to make the cost of both the turbine and generator prohibitive. During the last two years, however, with runners of increased specific speed, a much higher rotational speed can be secured for the same head and power, so that the turbine and generator designs have become thoroughly practical and economical.

Although it is true that the rotational speed secured in such a turbine is low when compared with the speed secured with turbines of the same specific speed and having two or more runners, there are many mechanical advantages involved in the use of the single-runner vertical turbine for low heads which more than overbalance the increased cost of the generator.

Many recent prominent low-head plants have adopted the single-runner vertical-shaft type of turbine. Had these plants been built 5 years ago, it would have been impossible to install vertical-shaft single-runner units of the same capacity, owing to the low speed and the resultant size of turbines and generators. For the same capacity, it would have been necessary to install turbines with more than one runner, or if single-runner vertical-shaft turbines were used, the capacity of each necessarily would have been considerably less than that of the turbines adopted, owing to the excessive dimensions with which it would have been necessary to deal.

132. Disadvantages of Multirunner Type. Now consider the disadvantages of the multirunner type of turbine.

(1) Two or more sets of gate mechanisms are required.

(2) Equal gate openings on all runners are difficult to obtain at all degrees of opening, owing to the lost motion or torsional deflection in the gate operating shafts.

(3) One or all of the gate mechanisms are completely submerged and are not accessible until the unit is shut down and the wheel casing is drained. Therefore, the breakage of a gate connection may be detected only by the falling off of the power output from the generator. In addition to this, a number of the main shaft bearings are inaccessible.

(4) When multirunner horizontal-shaft type units are installed, it is frequently necessary to have the generator floor below high tailwater level.

(5) It is often impossible to place the runners sufficiently far below the headwater surface to avoid the formation of vortices and the drawing of air into the turbines. This condition frequently impairs speed regulation as well as efficiency, and invites corrosion. When multiple-runner horizontal units are set in enclosed casings, this trouble may be avoided; but another disadvantage then appears, since there is a loss of head as the water passes each wheel and draft chest, and the runners last in line have in some cases been found to be operating under a considerably reduced head.

(6) In the case of a vertical turbine having more than one runner, the depth, and consequently the cost of substructure of the power house, is necessarily much greater than in the case of a vertical-shaft single-runner turbine.

(7) The cost of erection and of dismantling for repairs in the case of multirunner units of either the vertical or horizontal type is considerably more than in the case of the vertical-shaft single-runner type.

(8) The opportunity for lost motion in the connections between the governor mechanism and the wicket gates is, of course, greater the larger the number of runners used with the corresponding gate mechanisms. Consequently, in the case of multirunner wheels, there is a proportionally greater opportunity for hunting to develop in the operation of the governors than in the case of single-runner turbines.

(9) In any type of turbine having two or more runners in which the runners discharge against each other into a common draft chest or chests, there is considerable loss in efficiency due to interference of discharge as already explained, unless the distance between runners and, therefore, the length of the turbine and the cost thereof is greatly increased. The higher the specific speed, the higher is the discharge velocity from the runner buckets and, as the loss in the draft chest or tube is proportional to the square of the velocity, the loss in efficiency increases with the specific speed. It necessarily follows that in the case of high specific-speed wheels, great care must be exercised in properly designing the draft chests and draft tubes. The avoidance of sharp bends in the immediate neighborhood of the runner discharge is an advantage obtainable with the single-runner vertical turbine.

133. Advantages of Single-Runner Type. Among the principal advantages of the single-runner unit are the following:

(1) Only one gate mechanism is required, and this is located above the head cover of the turbine and is accessible at all times for inspection while the unit is in operation. The only parts of the turbine that are submerged are the runner and the guide vanes. Repairs can be made to the gate operating mechanism without dismantling the turbine.

(2) Owing to the fact that only one gate operating mechanism is used, involving a small number of parts, the chance for breakage is reduced to a minimum, and lost motion and deflection in the governor engine connections are avoided.

(3) It is possible to secure in a single-runner unit an ideal draft tube of long tapering section, without an obstruction or sudden

turn. Therefore, with this type of wheel it is possible to use runners of the very highest specific speed, as the draft tube can be designed to convert the velocity at the discharge from the runner buckets into effective head with small degree of loss.

(4) With a single-runner vertical unit, it is possible to mould in the concrete a spiral turbine casing similar in design to the cast-iron spiral casings used in connection with high-head turbines. It would be impracticable to prepare spiral casings for vertical or horizontal turbines having two or more runners, for obvious reasons. In a single-runner turbine operating in a spiral casing, the water is directed to the runner at uniform velocity around its entire circumference, producing more uniform operation and higher efficiency thereby. In the case of the multirunner vertical or horizontal wheels, however, it is necessary to set these wheels in open flumes or cylindrical casings, in which case the water is not guided uniformly to the runner, and, consequently, the approaching water is in a more or less turbulent state from eddies and whirls. In order to eliminate excessive loss in efficiency due to these eddies and whirls, it is essential to keep the velocities extremely low in the flume by increasing the dimensions of the flume, and therefore the distance between wheel centers.

(5) It is very often possible with turbines of the single-runner vertical-shaft type, to locate the runner and gate mechanism above high tailwater level, so that after the closing down of the head gates and drainage of the wheel pit, an attendant may examine all parts of the turbine without first having to pump out the wheel chamber.

The theory of the flow of water through runners as derived from low specific-speed wheels is not applicable, without modification, to the design of high specific-speed runners, on account of a number of new factors which must be considered. Low-speed wheels are such as are usually applied to high-head turbines. The runners installed in the various units at Niagara Falls are examples of low- and moderate-speed types.

134. Performance. Exact data regarding the performance of projected turbines of large size, when the specific speed is high, can be most easily and accurately obtained through tests on a model at the Holyoke flume.

It may be of interest to note that the experimental runner

tested at Holyoke for the two 17,000-horsepower turbines recently installed in the plant of the Pennsylvania Water & Power Company, at McCall Ferry, Pennsylvania, gave an efficiency of 90.62 per cent. The actual efficiency of the large units, as approximated from the power output, is in the neighborhood of 87.5 per cent, showing a loss of approximately 3 per cent. The McCall Ferry turbines are of the vertical-shaft two-runner type, each runner discharging downward into an independent draft tube, which is the best possible arrangement for a turbine of this type. It was necessary, however, in the case of the draft tube below the lower runner to make an extremely sharp turn so as to avoid excessive excavation, and both draft tubes were so modified on account of structural conditions that the distribution of areas is hydraulically defective. The turbines are installed in open flumes of rectangular dimensions. The difference in efficiency, therefore, between the model runner and the large turbines may be easily attributed to loss due to unfavorable hydraulic conditions in both draft tubes, and particularly to the sudden turn in the lower tube, and also to the usual losses which occur in an open-flume setting.

In the case of the Appalachian turbines, which are of the vertical single-runner type, having concrete spiral cases and ideal draft tubes, very exhaustive efficiency tests were made on two of the wheels, measuring the water by means of a carefully constructed weir 82 feet across the crest. The average efficiency secured in the two wheels tested was found to be 93.7 per cent. The model runner showed at Holyoke a maximum efficiency of 89.68 per cent. Consequently, the large units in place showed 4.02 per cent in excess of the maximum efficiency secured in the model runner.

This comparison clearly indicates that the vertical single-runner turbine with a spiral casing is more efficient than the experimental wheel, while the efficiency of the vertical two-runner turbine operating in an open flume is considerably below the efficiency of the experimental setting; and therefore the difference in efficiency between the vertical single-runner turbine and the two-runner vertical turbine is a very appreciable amount.

There have been quite a number of horizontal two-runner units, having central discharge chests, which have been carefully tested in place. These units seldom show efficiencies above the experi-

mental models. The only cases in which large turbines of this type have exceeded the efficiency of the experimental model have been in units of very high power having carefully designed draft chests, or in instances where the specific speed was not high; or where the experimental models used in the comparisons were tested in inefficient settings.

SPEED CONTROL

135. Chief Considerations. The problem of speed regulation may be considered in relation to two considerations which may exist in a plant: *first,* the control of speed during sudden changes in load which occur during normal operation; and *second,* the control during a period of constant load on the units.

In addition to considerations of efficiency, deterioration, and mechanical arrangements of turbines, the question of speed regulation is of great importance where turbines are employed for driving alternators. Speed regulation is obtained by means of a governor driven from the turbine shaft, consisting primarily of a governor head containing fly balls controlled by springs and connected through a pilot valve to the main governor valve. At normal speed the fly balls hold the governor valve in its mid-position, but for any change in speed the fly balls move the valve, admitting pressure through a suitable number of relay valves to an operating cylinder so that the turbine gates are adjusted to suit the new load and to obtain the normal speed.

Some of the chief considerations in connection with speed regulation are:

(1) *Sensitiveness of Governor Head.* This involves quick response to sudden changes of speed and also response to slight changes of speed.

(2) *Steadiness or Stability of Action.* This important quality involves freedom from unnecessary gate movements or movements occasioned by conditions other than changes of load on the unit. It is not only necessary for a governor to respond to changes of load, but it is equally important that it shall not act when there is no change of load. The continual motion and vibration of the turbine gate mechanism produced by unstable governors is a source of wear and rapid deterioration of the gate mechanism which in turn accentuates the lack of stability and is very objectionable. In older types

of governors this tendency was usually overcome by the use of a dashpot to damp such motions, but the use of a dashpot for this purpose renders the governor sluggish and impairs its action.

(3) *Power of Governor.* For large units, a large amount of power is required to actuate the turbine gates. This power may be obtained from a delicate governor head by the use of a series of relay valves controlled primarily by a small pilot valve attached to the governor head. In order to obtain sufficient fluid to move the operating piston, the main valve on the operating cylinder must be large and heavy; so that with a large unit either a considerable number of intermediate valves between the pilot valve and the main valve must be used, or else the governor head itself must be given a considerable amount of power. The same result could perhaps be accomplished by the use of high pressures in the governor system, but the use of such pressures has proved very unsatisfactory in practice and has been abandoned in favor of moderate pressures. Since time is required for sufficient fluid to flow through each valve to move the next valve in series, increase of power by such means is limited by the speed of action required. The best solution seems to be in the use of fly balls having a weight and power corresponding in order of magnitude to the turbine which they must control. Such increase in power in the fly balls involves no sacrifice in sensitiveness since the governor head is mechanically connected to the turbine shaft and is forced to respond immediately to any change in speed of the unit. The sensitiveness will, in fact, be increased by the use of heavy fly balls, since the retarding effect of friction can be made relatively less.

(4) *Reliability in Operation.* It is recognized by engineers of experience that delicate and complicated mechanisms should be avoided, and all machinery made to stand severe service and to be as nearly foolproof as possible. For this reason governors in important stations should be of as simple and rugged a design as possible. The effect on a large and heavy governor of accidental conditions, such as the presence of grit in the governor fluid or sticking of the governor parts, will be slight as compared with the effects on a delicate piece of apparatus.

(5) *Pumping System.* The pressure system used was formerly divided into a separate pumping equpment for each unit in a station.

This has been abandoned in favor of a single pumping system supplying the entire station. This change has greatly reduced the cost of attendance and has improved the continuity of service. The open system of governing is now being used, in which oil or water for the governors is pumped from an open tank and no pressures less than atmospheric are used at any point in the system. This change avoids troubles caused by air collecting in the pipes or pumps or the breakdown of oil under high vacuum.

(6) *Hand Control.* Another recent improvement is in the nature of the hand control used for the operation of the turbine gates. Originally the hand control of a turbine was accomplished by mechanical means through trains of gears. The mechanical efficiency of such a mechanism is so low that the time required to close the gates of a large turbine by such means is prohibitive. The mechanical hand gear was gradually replaced by a hand wheel incorporated in the design of the governor, controlling the turbine gates by oil pressure through the main valve of the governor. This arrangement, however, although it has been recently used by a number of the standard governor builders, is subject to the objection that, should the governor valves be dismantled for repairs, the turbine must be shut down, and it is poor engineering to dispense with the services of the turbine for the sake of a governor valve. The best method of hand control now adopted for large units, seems to be in a separate device, entirely independent of the governor and placed at a convenient point, and so designed as to admit oil from the central pumping system directly to the turbine operating engine. A restoring-mechanism connection from the turbine gates maintains these gates in a position corresponding to the position of the handwheel. By this means it is possible to move by hand the gates of the turbine through their entire range in approximately 10 seconds of time. Except for units of very large size, a small hand operated plunger pump may be supplied in connection with the operating stand, by means of which it is possible to create pressure for the movement of the gates in the event that the central pumping system is temporarily out of service.

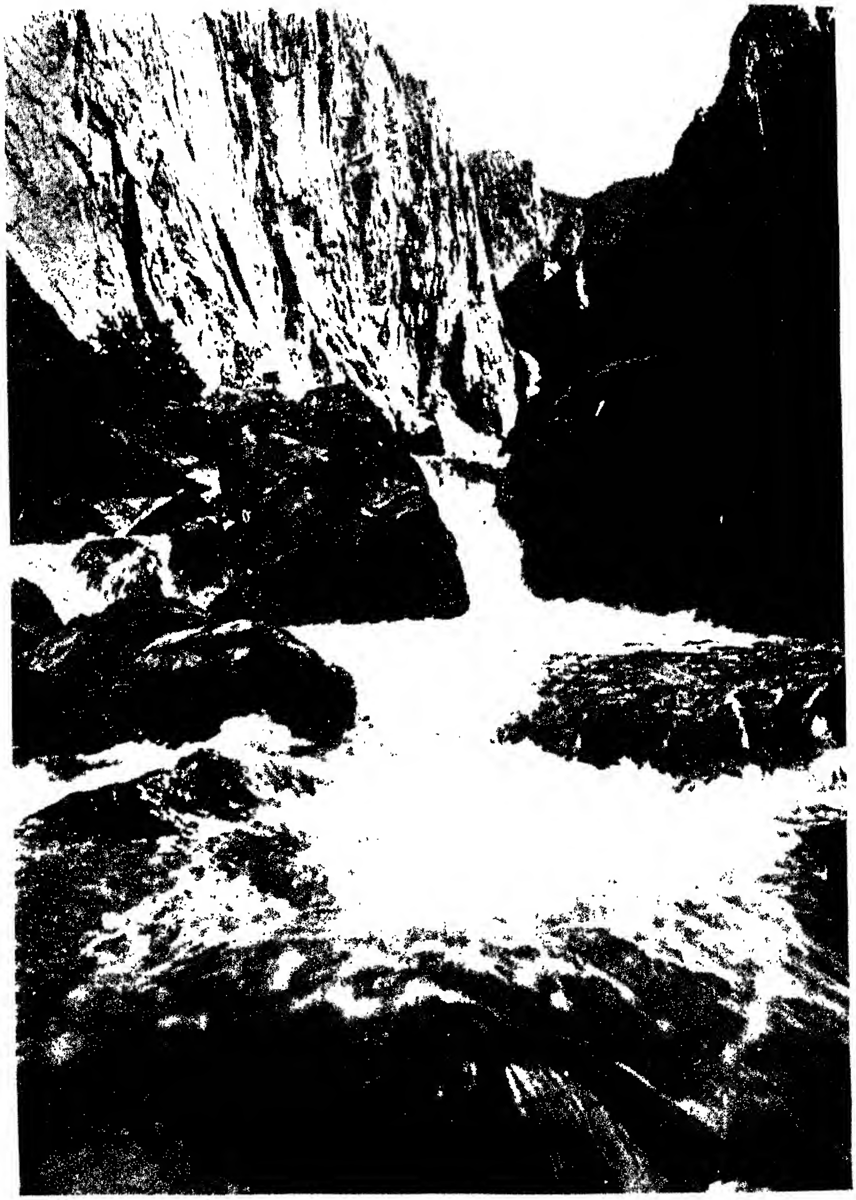
(7) *Governor Mechanism.* The mechanism of the modern governor itself has been reduced to a few comparatively simple operations. Certain compensating devices are required to restore the

governor valves after the required gate movement has been produced and also to restore the speed of the turbine to the required value after the water column in the penstock, turbine casing, and draft tube has had time to be accelerated or retarded to its new value. The governor should also have adjustments so that the change in speed from full-load to no-load conditions may be fixed to suit the parallel operation of alternators and to adjust the time of action of the governor to suit the length of and velocities in the water passages, also an attachment for changing the normal speed of the turbine. Modern governors are also fitted with motors by means of which the speed of the turbine can be adjusted from a distant point, such as the switchboard, this device being used in synchronizing the unit.

136. Other Factors. The above points cover all the features which can be controlled in the design of the governor itself. The speed regulation of a unit, however, depends on several other factors which are invariably of greater importance than the action of the governor itself; these are the length of and velocities in the penstock and draft tube, as well as the length of the water column in the turbine casing itself; also the "flywheel effect" of the rotating masses of the generator and turbine.

A perfect governor will be unable to produce a speed regulation better than that permitted by the factors just mentioned. These factors are controlled by the design of the power development as a whole, so that the actual speed regulation obtained is only affected to a limited extent by the construction of the governor itself. Properly, all of these factors should be considered together.

In a majority of the large installations recently undertaken, turbine governors have been designed and constructed by the builders of the turbines, thus producing a governor suited to the exact conditions of each turbine and avoiding the division of responsibility which has so often proved to be a source of annoyance. The better appreciation of the factors entering into governor problems has resulted in greatly improved speed regulation which, when taken in connection with valuable improvements in the mechanical design of the governor, has removed this important auxiliary from the class of necessary evils and placed it in that of reliable machinery.



VIEW OF HEADWATERS USED IN GUNNISON TUNNEL PROJECT
Photo by Brown Brothers, New York City

WATER-POWER DEVELOPMENT

PART III

CONDUCTION OF WATER

CONTROL

137. Headrace and Tailrace. Races should be designed in accordance with the laws of Hydraulics to insure as small a loss of head as possible. Protection from freezing requires the races to be comparatively narrow and deep rather than wide and shallow, and also that the velocity of flow be maintained at not less than 2 to 3 feet per second.

Where conditions require, there should be placed across the entrance to the headrace, either a boom, or cribs with submerged openings, or submerged masonry arches, in order to prevent the passage of ice or floating dêbris. That alignment should be chosen which will not only prevent local accumulation, but will deflect the dêbris downstream.

Where much sand is carried in suspension, the location and direction of the entrance to the headrace must be carefully selected, and one or more enlargements of sections of the headrace should be provided, with the object of reducing the velocity of flow so as to encourage the deposition of the sand at these sections. To check the motion of sand which may tend to roll along the bottom, narrow upright boards or other obstructions are sometimes built across the line of flow on the bed. In all such cases proper provision must be made for flushing out or otherwise getting rid of the accumulated sand from time to time. Near the lower end of the headrace a boom should be placed to direct to the sluice gate of a wasteway the ice and other floating matter which may pass into the race.

138. Water Racks. These may be single or double, the latter consisting of a coarse rack in front of a fine rack. The object of these racks is to screen out of the water such floating or suspended matter as might choke or damage the wheels. In fine racks the

clear space may be $\frac{3}{4}$ inch to $1\frac{1}{2}$ inches; in coarse racks about 3 inches. The clear space between bars of a rack should be less than the smallest opening of the water passage in the guide or runner buckets; and, to permit free flow against the frictional resistance offered by the racks as well as at times of partial clogging of racks, there should be a considerable excess area of total clear space between the bars. Inclined racks offer a larger wetted area for a given depth of submergence than vertical racks, and are more readily cleaned. For practical convenience, racks are often constructed in sections which are easily removable.

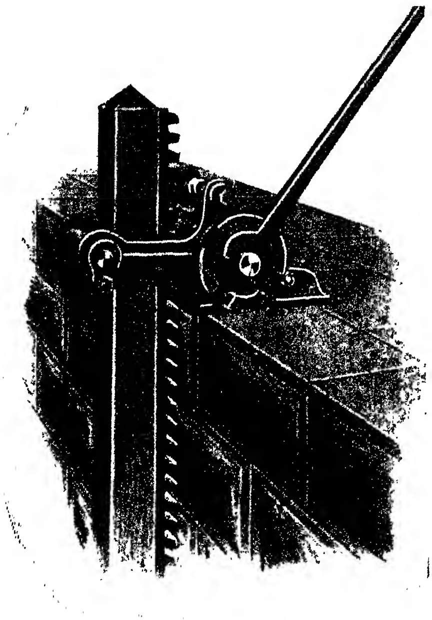


Fig. 142. Typical Head-Gate Hoist
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

***139. Head Gate.** The purpose of the head gate, Figs. 142 to 153, is to control or shut off the water from a penstock, flume, open turbine chamber, or forebay. It is usually a vertical gate sliding in guides, and, until quite recently, was constructed of wood held together and braced by iron bolts and straps. Head gates are now frequently built up of steel plate and structural steel—more particularly those of large size. Such gates are actuated by hand, by means of rack-and-pinion or screw spindles, or by special devices operating under mechanical electric, or hydraulic power. They are frequently counterweighted; and for large gates a by-pass or balance port is usually employed, by means of which the water pressure on the two faces may be balanced before moving the gate. In some cases, friction roller bearings are employed to reduce the friction; and in others, by an ingenious contrivance, the

*Articles 139 to 141, inclusive, are adapted from "Modern Turbine Practice", John Wolf Thurso, D. Van Nostrand Co., New York.

gate is lifted from its seat in the preliminary action of opening. In order to reduce the wear, the gate is sometimes designed to slide on special guide bearings, instead of on its seat, which it does not touch until reaching its position of complete closure.

A cylindrical gate, built up of plate and structural steel is shown in Fig. 151. The gate is so pivoted that it is practically balanced so as to make operation easy.

Another form of head gate which has not come into general use, though possessing many positive advantages, consists of a cast-iron

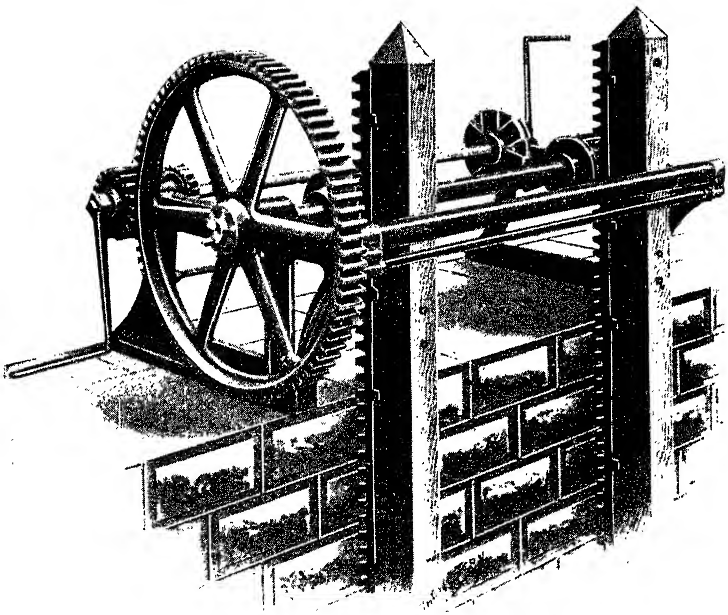


Fig. 143. Compound-Gear Head-Gate Hoist for Double-Stem Gate
Shafts carrying spur gears and pinion are mounted on cast-iron stand; wrought iron cranks are used for operating the hoist.
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

cylinder, double seated, the lower seat being formed by a ring attached to the floor of the headrace, and the upper seat by the edge or rim of a dome forming the head, which is fastened by steel rods to the lower ring, as in Fig. 152. The cylinder itself is raised or lowered by means of a chain. In the center of the dome is a small filling gate operated by a separate chain. It is thus practically balanced, and hence requires but little power in operation; more-

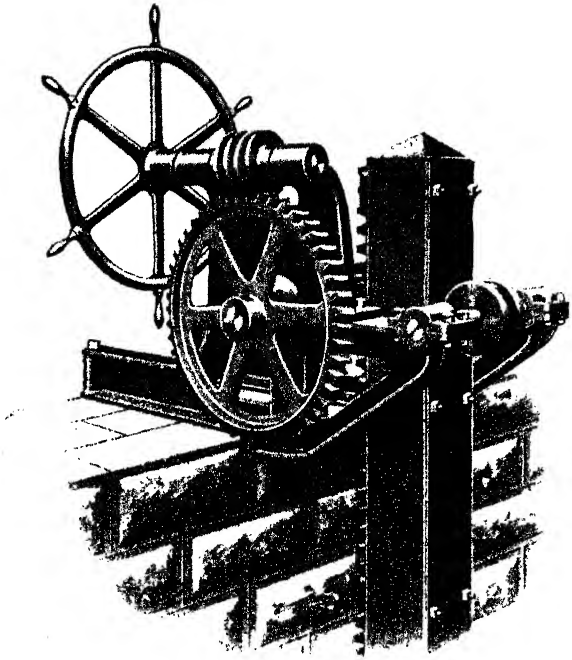


Fig. 144. Single-Stem Head-Gate Hoist, with Pilot Wheel and Worm
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

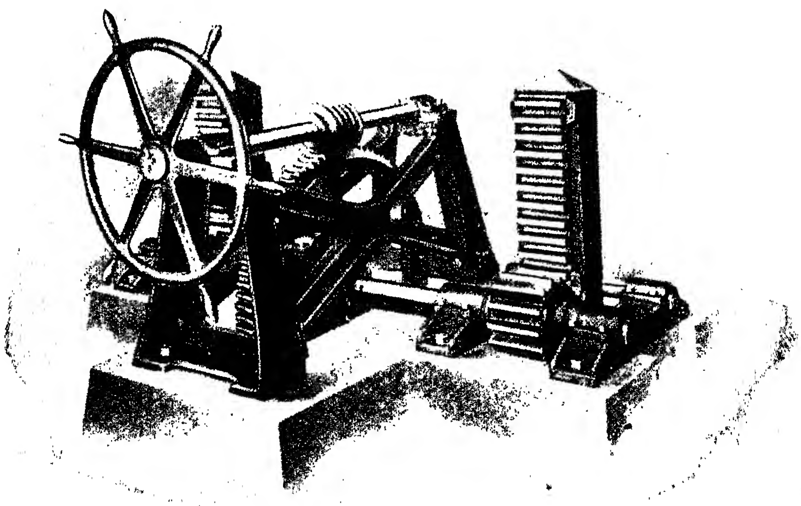


Fig. 145. Double-Stem Worm Wheel Hoist
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

over, the lift required to give a clear waterway equal in area to that of the circular opening which it controls, is but one-quarter of the diameter of such opening.

Where the necessary head room is not available, wicket gates turning on a vertical spindle may be used, as in Fig. 153. Such a gate requires less power to operate than a sliding gate, but is generally liable to greater leakage. When open, it presents its edge to the current, and so offers some, though no great obstruction to the current.

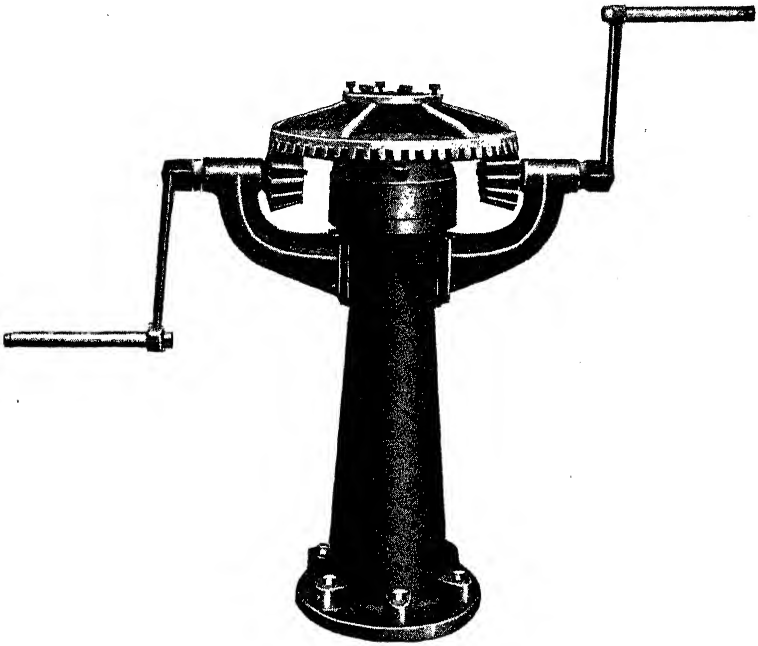


Fig. 146. Head-Gate Hoist for Operating Gates under Pressure
Upper end of gate is threaded to match a bronze nut attached to bevel gear shown in cut. Ball bearings are used to insure easy operation.
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

It is advisable to design gates which are intended to be used in cold climates, so that they may be lowered entirely below the surface ice when closed, and may be raised entirely clear of it when open.

140. Gate House. In many important water-power projects, the flow into the canal is controlled by a series of gates, with their hoisting gear and appurtenances all enclosed in a covered building called a gate house.

CONDUCTORS

PENSTOCK

141. Function. The term *penstock* is applied to the pipe which brings the water from the canal or other source of supply, to the turbine chamber. When the source of supply is near the motor,

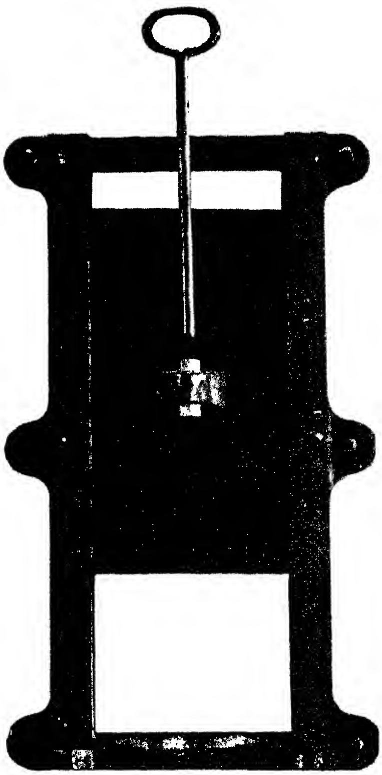


Fig. 147. Filler Gate, with Frame for Bolting to Timber Head Gate

Allows passage of sufficient water to equalize pressure, so that head gate need not be lifted under total pressure.

Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

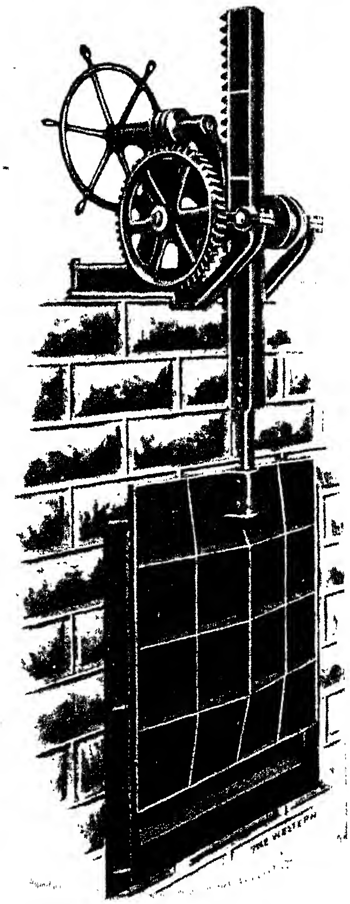


Fig. 148. Square Cast-Iron Head Gate with Hoist

Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

it is a relatively unimportant detail of the system. On the other hand, it sometimes happens that this pipe is several miles long; in which case it assumes a position of primary importance; in fact it may become one of the controlling features or elements of the design.

Penstocks, or feeder pipes, may be made of riveted wrought iron or steel, of wooden staves, or of concrete steel. The design of penstocks will not be considered in this work. They should always be as short as possible, even when a shorter penstock involves a

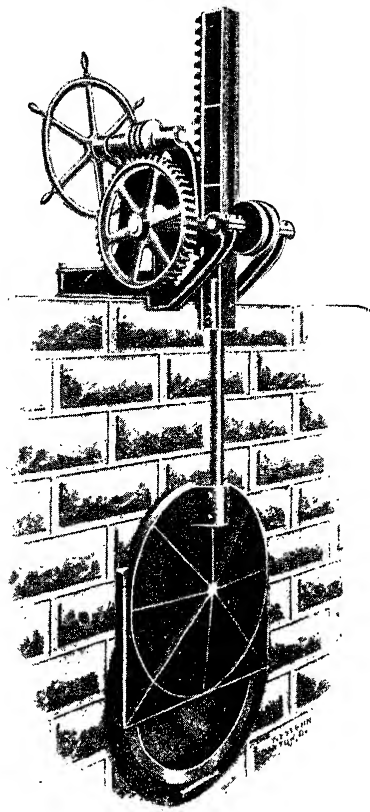


Fig. 149. Circular Cast-Iron Head Gate with Hoist
Courtesy of Dayton Globe Iron Works Company, Dayton, Ohio

greater expenditure for excavation, etc. The shorter the penstock, the better it is for the speed regulation of the turbines, and the less steel-plate work has to be kept painted and repaired.

142. Speed of Water. The following rules should be observed when determining the cross-sectional area of the conductors which convey the water to and from the turbines:

"The speed of the water should be gradually increased from the speed in the headrace, usually 2 or 3 feet per second, to the penstock

speed, by means of a cone or taper piece, as in Fig. 154. Near the lower end of the penstock, the speed should again be gradually increased, so that the water will arrive at the guide buckets with a speed equal to that with which it has to enter these guide buckets. At the entrance of the draft tube, or draft-tube elbow or tee, the water should have a speed equal to the absolute velocity with which it leaves the runner buckets, and should then gradually decrease to a speed of about 2 or 3 feet at the lower end of the draft tube. A speed of 2 or 3 feet also is chosen usually for the tailrace.

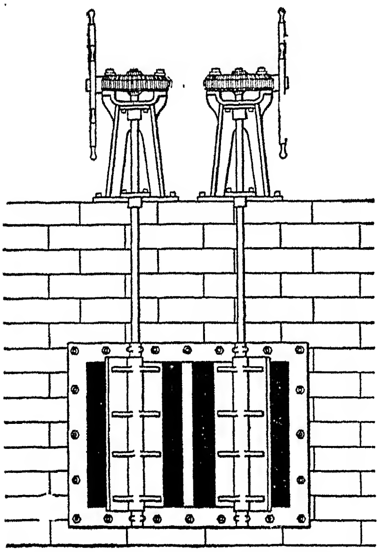


Fig. 150. Worm-Geared Swing Gate

"In general it should be stated: Avoid changes of speed of the water where possible; but

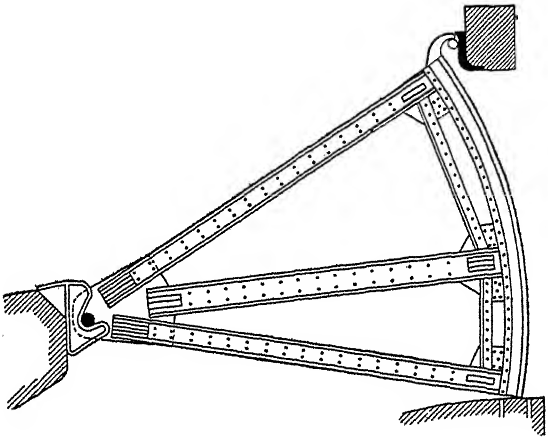


Fig. 151. Cylindrical Balanced Gate Built Up of Plate and Structural Steel

where such changes are necessary, make them gradually; also, avoid changes of direction of water; but where such changes are necessary, use curves of long radii.

"The arrangement often employed, of having at the lower end of the penstock, and at right angles to the same, a drum or receiver of much larger diameter than the penstock itself, from which drum a number of turbines are supplied by branches set at right angles to the drum, must be condemned on account of the resulting abrupt changes in speed and direction of the water.

"All nozzles or branches of penstocks should be at an angle of not over 45 degrees to the penstock; or, in other words, the directions of flow of the water in the penstock and in the nozzle or branch should form an angle of not over 45 degrees with each other.

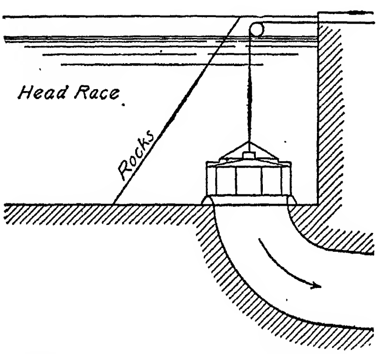


Fig. 152. Cast-Iron Cylindrical Double-Seated Head Gate

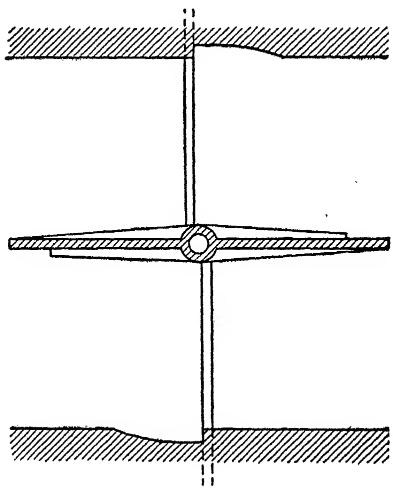


Fig. 153. Wicker Gate Turning on Vertical Spindle

Directly beyond each nozzle or branch, the diameter of the penstock should be reduced, to keep the speed of the water uniform.

"When determining the water speed in the penstock, all conditions should be carefully considered; it should be remembered that the friction loss in a penstock varies with the square of the speed.

"Conditions making a low speed advisable are: low head; large diameter of penstock; great length of penstock; many bends in penstock; variable loads on the turbines; regulation of speed of turbines by changing the amount of water used.

"Conditions making a high speed permissible are: high heads; small diameter of penstock; short penstock; few or no bends in penstock; steady loads on the turbines; regulation of speed of turbines by by-pass.

"Many hydraulic engineers employ in all cases a penstock speed of 3 feet per second; but it is often of an advantage greatly to exceed

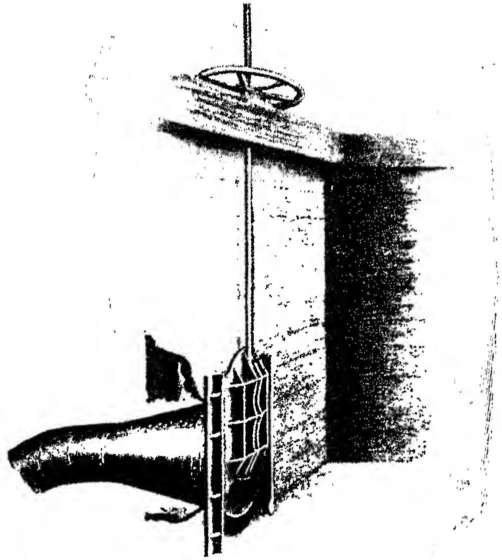


Fig. 154. Entrance Taper and Head Gate in Flume
Courtesy of Pelton Water Wheel Company, San Francisco, California

this velocity. From a great number of well-designed water-power plants constructed in America and Europe during recent years, the writer has deduced the following tabulation of highest permissible

Speeds of Water Permissible in Penstocks

Penstock Diameter (ft.)	Water Speed (ft. per sec.)
4	12
5	11.5
6	11
7	10.5
8	10
9	9.5
10	9
11	8.5
12	8

speeds of water in penstocks of a length of 1000 feet or less, with easy bends, and provided with proper arrangements for the protection of the penstocks against water hammer:

"In penstocks of 1 or 2 feet diameter, speeds as high as 20 to 30 feet have been used. With very low heads, the penstock speed is often limited by the amount of head that it is permissible to lose in the penstock.

143. Losses of Head. "The principal losses in the head of the water while entering the penstock and flowing through the penstock and draft tube, are due to the following causes:

"(1) *Entrance.* This loss may be kept low by having a large entrance connected to the penstock by an easy cone or taper piece. With the usual head-gate arrangement, such large entrance openings require very heavy and cumbersome gates for penstocks of large diameter; but there is no reason why this taper piece could not be partly or wholly in front of the gate and inside the headrace or forebay. The penstock entrance should always be as much below the surface of the water as circumstances will permit.

"(2) *Friction.* This loss may be kept down by a low speed of water, and by smooth interior of the penstock and draft tube.

"(3) *Changes in Direction of Flow.* This loss may be kept down by using as few and as easy bends as possible.

"(4) *Changes in Speed of Water.* This loss is due to the conversion of part of the energy in the water into another form, and may be kept low by having as few and as gradual changes as possible.

"(5) *Speed of Water Leaving Lower End of Draft Tube.* This loss is equal to the velocity head, corresponding to the speed with which the water leaves the draft tube, and may be kept down by making this speed low.

144. Protective Devices. *Safety Head.* "Long penstocks, carrying water at high speed, should be provided with a safety head, besides the usual devices for the protection of the penstock against water hammer. For this purpose, a cast-iron or angle-bar flange is riveted to the lower end of the penstock, to which flange the head closing the lower end is bolted. The flange bolts should have a factor of safety of not more than about half the factor employed for the rest of the penstock. Between flange and head a packing of dry white pine should be used, which, when water is admitted to the penstock, swells and makes a tight joint. Where the end cannot be used for this purpose, large nozzles may be riveted to the penstock, located as nearly as possible in the line of the water

hammer, and closed by heads secured as already described. The end of the penstock, or the nozzles, should be so situated that, should the heads blow out, no damage will be done by the jet of water issuing from the opening. This arrangement will not only save the penstock and turbines from being wrecked in case of severe water hammer, but also the power house from being demolished by the water set free.

Air Inlet. "Ample air inlets should be provided at the upper end of the penstock, as otherwise—should the safety head by some chance give way, or the turbine gates or turbine stop valves, if such are employed, be opened while the head gate is closed, and the penstock is full of water—the penstock might collapse by the vacuum created in its interior. Care must be taken to prevent the water in the vents or air inlets from freezing, as this would render them useless.

Standpipe. "A penstock which is carried for a considerable distance at about the same elevation as that of its inlet, and with so little slope as to be nearly horizontal, and then descends to the power house on a steep grade, is liable to collapse when the turbine gates are opened quickly, as the water in the inclined part has the tendency to increase its speed more quickly than the water in the horizontal part, and may thus break away from the latter and cause a vacuum in the penstock. An air-inlet valve will prevent this, but it is better to have a small compensating or equalizing reservoir (or standpipe) at the junction of the horizontal and inclined part of the penstock. Such a reservoir may be built of steel plate, concrete, or masonry, and will not only prevent the collapse of the penstock from the cause above named, but will also greatly improve the regulation of the turbines, and decrease the water hammer in the penstock, acting, in fact, in the same manner as a standpipe.

Expansion Joint. "Expansion joints in steel penstocks are not as important as is often asserted, since most penstocks contain bends which permit of a limited movement large enough to compensate for expansion and contraction; but in a straight steel penstock rigidly held at each end, the strains due to changes in temperature are very heavy, and in such case expansion joints must be provided.

"The lower end of a penstock should be held very securely in all cases, to prevent forces due to temperature changes and other causes

from throwing the turbines out of alignment, cracking the power-house walls, etc.

145. General Construction. "Steel-plate penstocks are usually made in small and large courses, and lap riveted. Butt-strap joints, with a single butt strap on the outside, offer less frictional resistance to the flow of the water, but are more expensive. A manhole should be provided at the upper end of a penstock, and at the lower end also, if required. When repainting the inside of a penstock, or repairing the same, the water leaking through the head gate should be prevented from running down the penstock; and for this purpose a small outlet nozzle, about 6 inches in diameter, and closed by a blank flange, is provided at the lower side of the upper end of the penstock; and by building a small dam of clay in the penstock, just beyond this nozzle, the leakage is prevented from flowing down the penstock. All openings in penstocks for large nozzles, branches, manholes, etc., should be reinforced by steel-plate rings riveted around the openings, to make up for the material cut away by the opening.

"Penstocks should be calked both inside and outside, and the plates thoroughly cleaned by scrapers and wire brushes before painting.

"Masonry piers are often damaged by the expansion and contraction of the penstock they support, and the paint is rubbed off the penstock where it rests on the piers. Such unprotected places are hidden from view by the masonry, and are apt to corrode very quickly, as water is always retained between the surfaces of contact of the masonry and the penstock. It is therefore preferable to use steel piers on concrete or masonry bases. Such steel piers are cheaper than concrete or masonry piers; they leave every part of the penstock accessible for painting and repairs, and are free to swing on their bases, like inverted pendulums, to accommodate themselves to any movements of the penstock caused by changes in temperature. The uprights or posts of these piers are provided with bolt holes, to fasten to them the studs for a housing over and around the penstock when desired.

"Except where the distance between the penstock and the ground varies considerably, the steel piers are all made the same; and the variations in the height of the penstock above the rock or

solid ground are made up in the height of the concrete or masonry bases. The uprights of the steel piers are anchored to the bases, or, if the latter are of small height, through the bases to the rock below.

"A penstock running down a steep mountainside must be prevented from sliding down the slope. Where concrete or masonry piers are employed, it is often sufficient to rivet short pieces of heavy angle bars to the penstock, and to have these bars bear against the uphill side of the piers; but with steel piers the penstock must be anchored to the rock or to special anchor piers. It is well to have, in any case, a specially heavy concrete or masonry pier at the lower end of the penstock, to prevent the latter from throwing the turbines out of alignment.

"In a climate like that of the northern part of the United States and of Canada, penstocks must be covered or boxed in, to protect them from the extreme cold; otherwise ice will form on their inner surfaces.

"During midsummer, the heat of the sun's rays acting on an empty penstock will often injure the paint, cause it to blister off, and perhaps overstrain the penstock itself; and a covering will therefore prove an advantage both in cold and in hot weather. Even in a well-protected penstock, ice will be formed in severe weather, when the water in it is allowed to remain stationary for more than a few hours at a time.

"Where the ground under a penstock consists of earth, it is preferable to bury the penstock below the frost line, like the water mains in a city street.

"A buried penstock is free from the bending strains occasioned in a penstock carried on piers by reason of the length between the piers being unsupported; but a buried penstock of large diameter will require stiffening angles to be riveted to the upper half of its circumference, to prevent it from collapsing by the weight of the earth above it.

"The cost of burying a penstock will be about the same as when masonry piers are used.

"Under the penstock, in the center of the ditch, should be a drainage ditch about 1 foot square in cross-section, and filled with pebbles or broken stone, as used for concrete making. The penstock should rest on short wooden blocks; and the main ditch should be

left open during the first year, or for one winter season at least; after which the penstock is carefully inspected, recalked where leaky, and repainted inside and outside; and then the earth is packed under and around the penstock, and the ditch filled in, the wooden blocks being removed as the work proceeds.

146. Stavepipe and Other Types. Wooden penstocks or stavepipes deserve a wider application than they have so far found in the Eastern States. Wooden penstocks are cheaper and will last longer than steel penstocks, need less protection against extremes

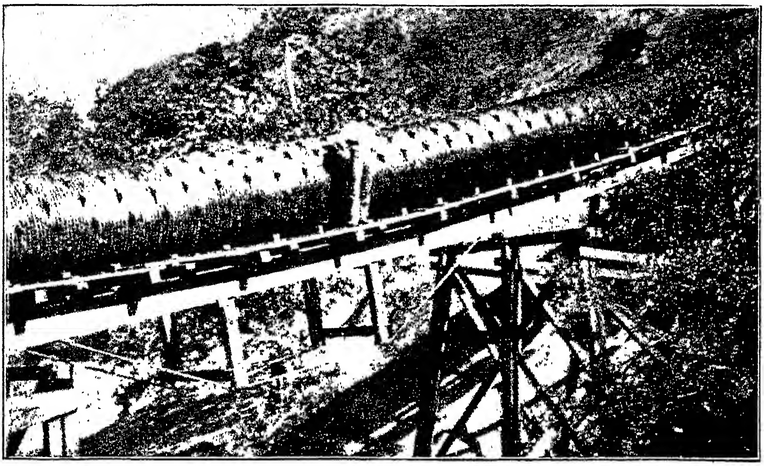


Fig. 155. Redwood Stavepipe under 100-Foot Head, Santa Ana Canal, California

in temperature, and require no painting. Their interior surfaces are smoother than those of steel penstocks, and therefore offer less resistance to the flow of the water. Such pipes have been built up to 9 feet diameter.

"Wooden penstocks are made of staves from 2 to 4 inches thick and from 6 to 8 inches wide, planed to the proper shape, and held together by round or oval iron or steel rods, connected by hoop locks of various designs. The staves must be thick enough, or the hoops spaced closely enough, to prevent the staves from bulging out between the hoops. Thick staves are usually provided on one edge with a bead of from $\frac{1}{16}$ to $\frac{1}{8}$ inch in height, by $\frac{1}{2}$ to $\frac{3}{4}$ inch in width, located next to the inner side of the stave, as with such a bead it will require less strain in the hoops to make the penstock water-tight.

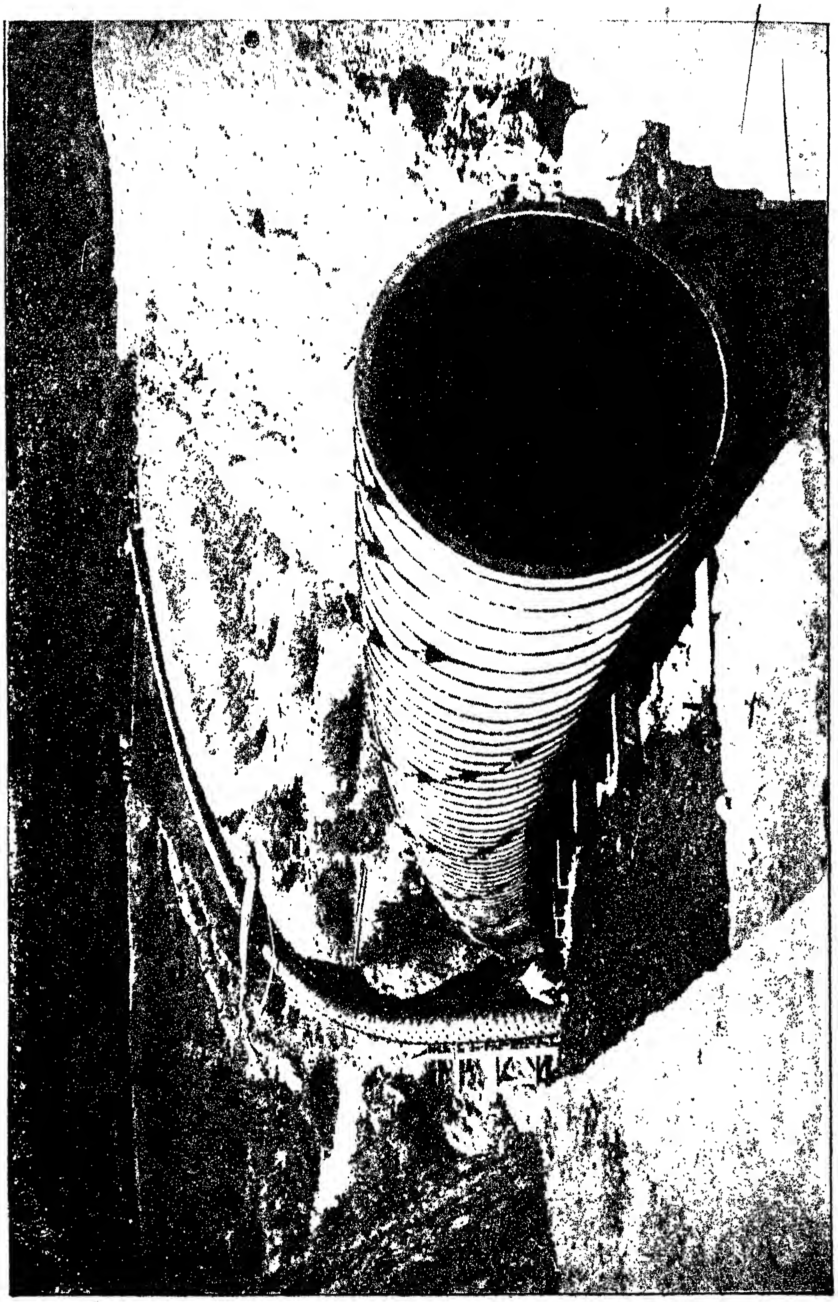


Fig. 156. Redwood Stavepipe (52-inch) Crossing Warmsprings Canyon, Near Redlands, California
Courtesy of U. S. Geological Survey

"The joints at the ends of the staves are usually made by steel tongues driven into kerfs. These joints must be well broken.

"Curves in wooden penstocks require a long radius, and therefore their horizontal and vertical alignment must be located on the ground, like a railroad line. The minimum radius, in feet, that can be used in a wooden penstock is about $R=12.5\times D_p\times t_s$, in which D_p is the inside diameter of the penstock in feet, and t_s the thickness of the staves in inches. Where a smaller radius is required, a section of steel penstock has to be inserted in the wooden one for the purpose.

"The wood employed should be clear and sound, and free from pitch, so that the staves will become saturated by the water. The wood used for such stavepipes is, in the order of its value for the purpose: California redwood, Douglas spruce (also called Douglas fir), spruce, white pine, southern pine, and cypress.

"The staves of a wooden penstock that is not left empty long enough to allow the wood to dry, will last much longer than the hoops; and the hoops may be renewed, when destroyed or weakened by rust, by placing new ones between the old hoops, if the soundness of the staves will warrant it. A stop valve should be used at the lower end of a wooden penstock, with no head gate at the upper end, to insure the penstock being always full of water, which may be shut out by the use of stop logs in case of necessity. A wooden penstock buried in the ground may be left empty for some time, without danger of the staves drying out.

"In heads of 200 feet and more in height, wooden penstocks are not economical, as the hoops require as much metal as the plates for a steel penstock.

"Penstocks constructed of concrete and steel also deserve a wide application, and should outlast both the steel and wooden penstocks, as the steel rods are protected by the concrete.

"Instead of welding together the ends of the embedded hoops, these ends may be run past each other for a distance of from 30 to 40 times the diameter of the hoop rod; or an inch or so of each end of the hoop rod may be bent back flat on itself, and the ends run past each other for a distance of from 20 to 30 times the diameter of the hoop rod. For small concrete penstocks, steel wire wound spirally can be used to form the hoops.

"For heads of 200 feet and more in height, penstocks built of

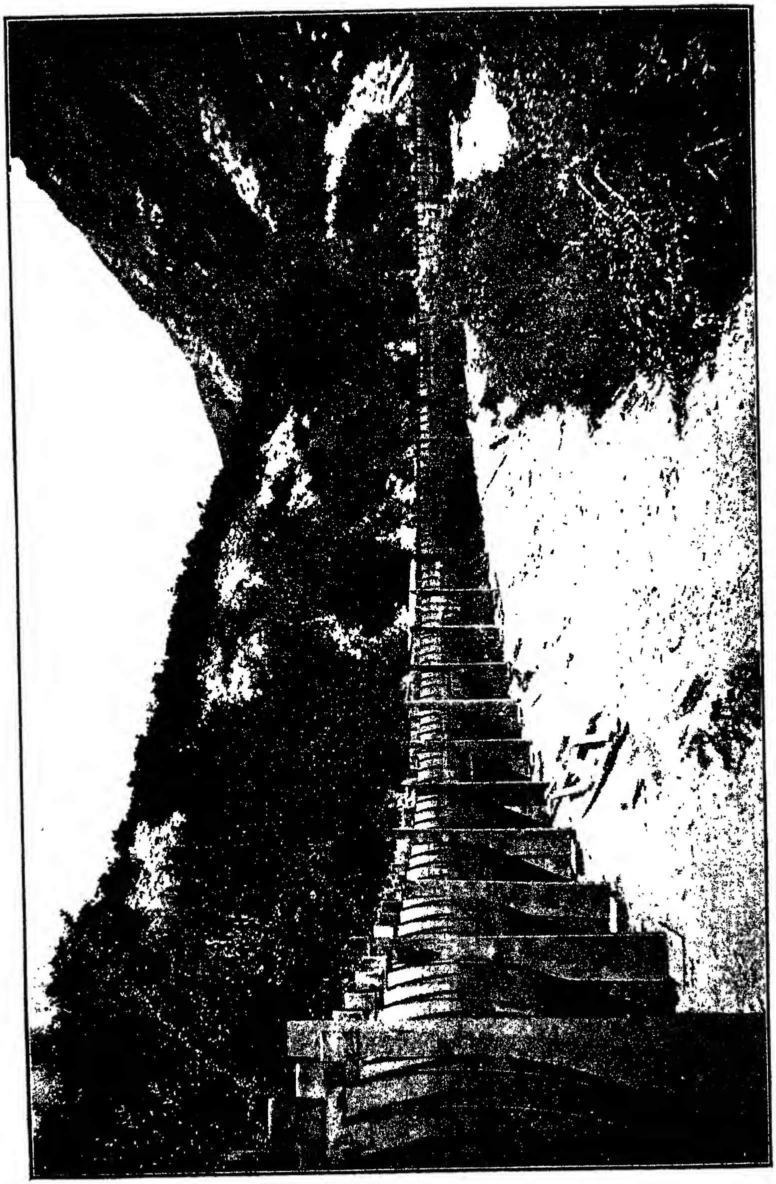


Fig. 157. Side View of Sterling Flume in Provo Canyon, Utah
Courtesy of U. S. Geological Survey

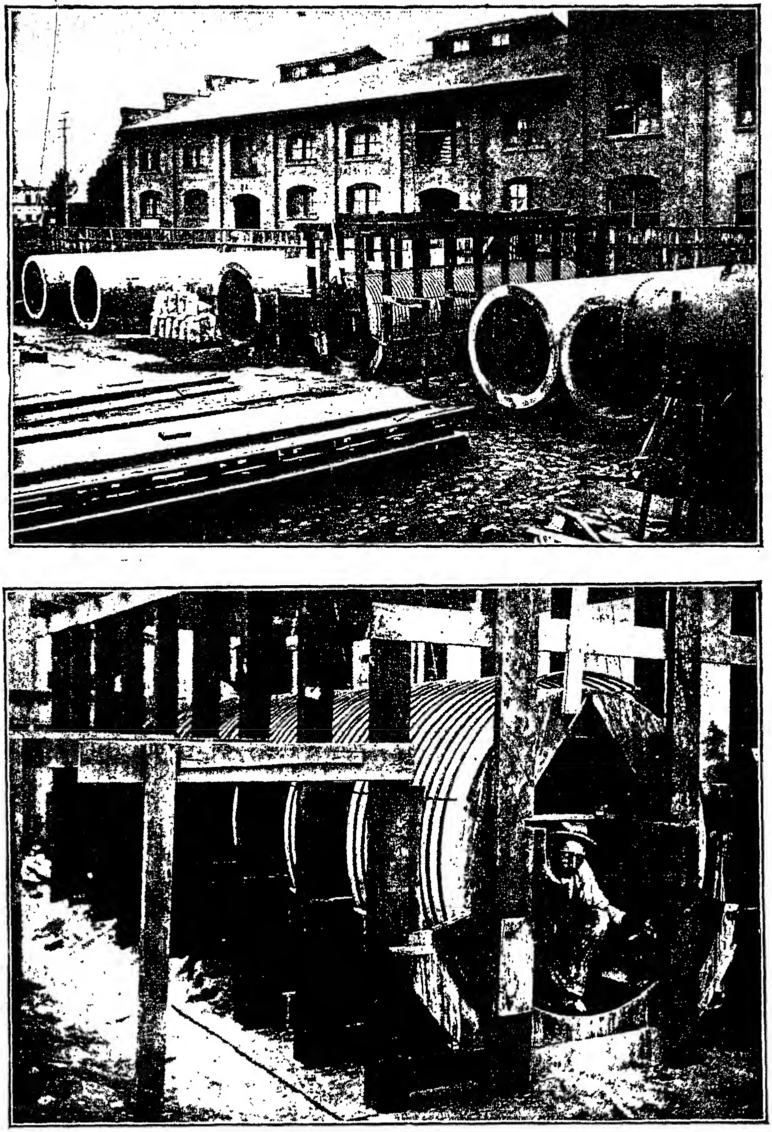
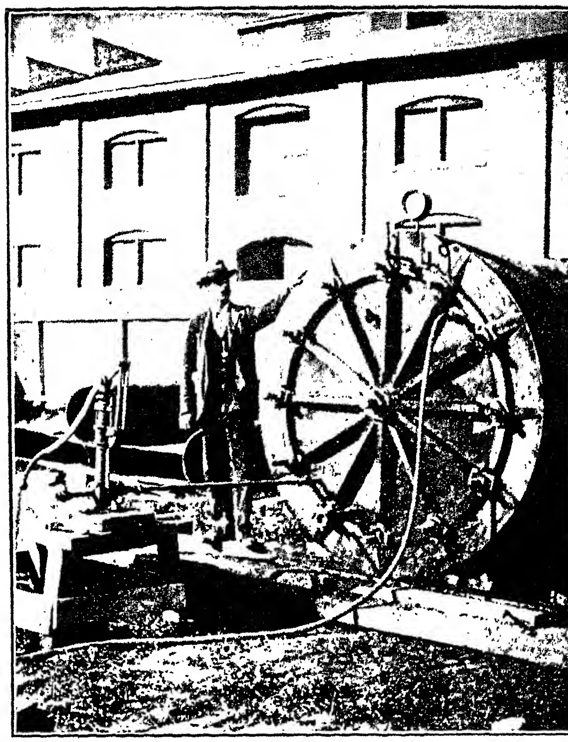
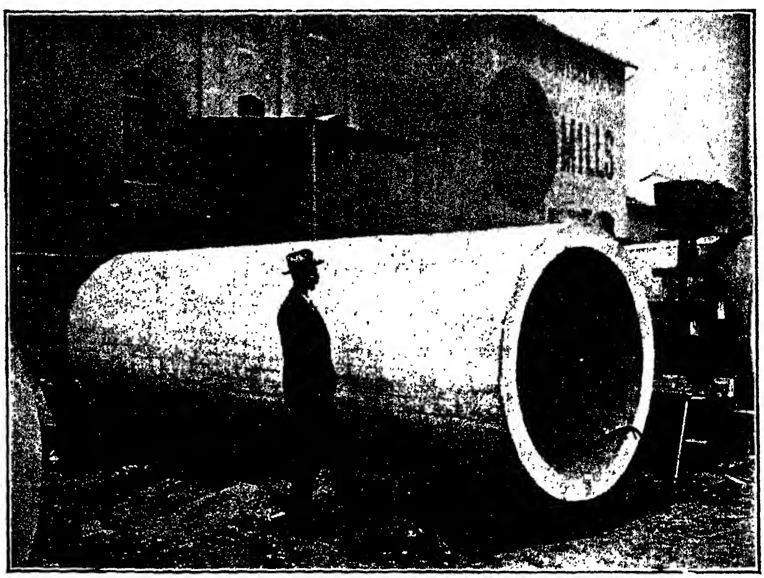


Fig. 158. Reinforced-Concrete Pipes as Made and Tested by U. S. Geological Survey



(A) Pipe under Test



(B) Method of Raising End of Pipe

Fig. 159. Reinforced-Concrete Pipes as Made and Tested by U. S. Geological Survey

concrete and steel are not economical, as the hoops require as much metal as the plates for a steel penstock.

"Standpipes may be built either of steel plate or of concrete and steel. An excellent arrangement is to have a concrete base straddling the penstock, and the standpipe placed on top of this base, like a steel chimney or stock."

Figs. 155, 156, and 157 are illustrations of wood stavepipes.

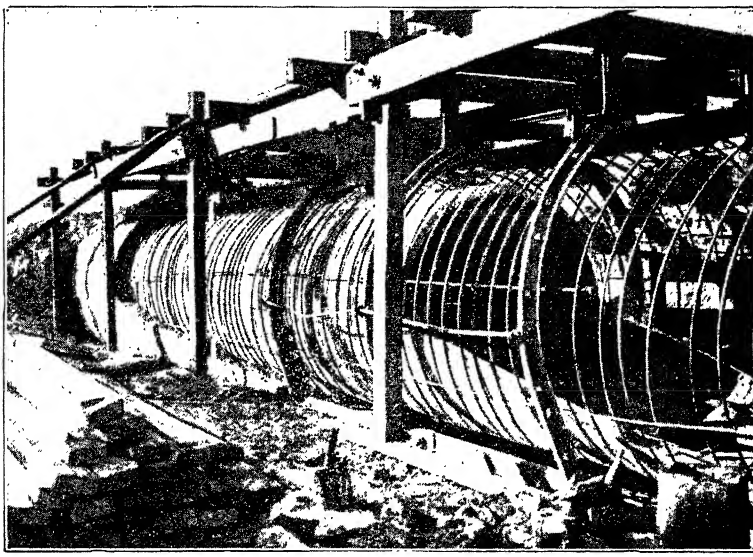


Fig. 160. Construction of Pipe by Movable Form
Courtesy of U. S. Geological Survey

Figs. 158, 159, and 160 illustrate steel concrete pipes as made and tested by the United States Geological Survey.

Riveted-steel pipes are shown in several of the illustrations of power plants appearing in subsequent articles.

TRANSMISSION

147. Methods. The older method of applying the power of water to industrial purposes consisted in conducting the water by means of canals or flumes to the various establishments in which the power was to be utilized. This method required special physical conditions of a very favorable nature, in order that great expense in construction might be avoided. The later method is to convert the

(A) Sixth section looking upstream

(B) Lining in rock cut, seventh section looking downstream

(C) Ninth section, rounding Wadsworth Point at mouth of canyon

(D) Junction of earth and concrete-lined section, showing warped surface in latter

Fig. 161. Concrete-Lined Section of Truckee Canal, Nevada

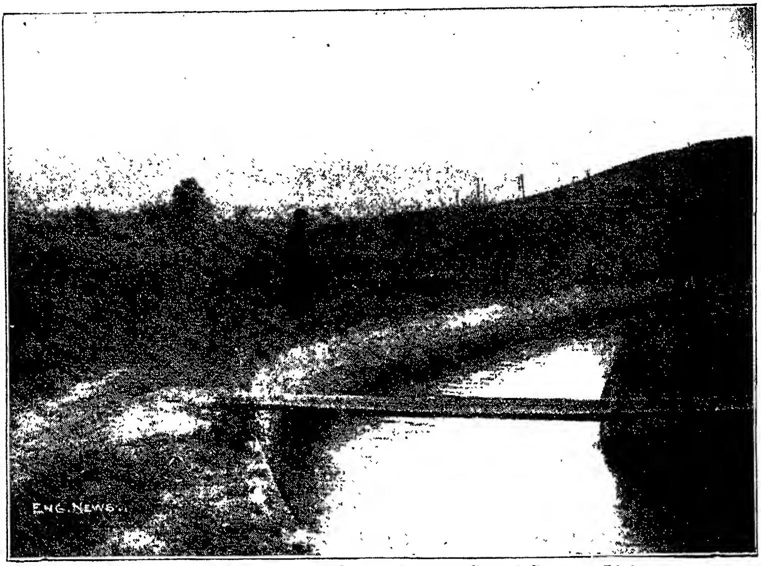


Fig. 162. Santa Ana Canal, California, with Gravel Concrete Lining

Fig. 163. Riverside Canal, California, Showing Sand Deposit Left on Bottom after a Year's Service

Courtesy of "Engineering News"

energy of the water into mechanical power, and the latter into a form suitable for transmission, in a single powerhouse, from which central station the power may be conveyed long distances, and applied to various machines located to the best advantage without reference to the waterfall itself.

Shafting, wire rope, compressed air, and water under pressure have all been more or less utilized for the transmission of power;

Fig. 164. Bear Valley Canal, Redlands, California
Boulder-Lined and Plastered
Courtesy of "Engineering News"

but, since the great development of electricity in comparatively recent times, this agent is most generally employed for that purpose.

148. Canals and Flumes. It does not usually happen that a fall available for power production (either in natural fall, or one created by damming a stream) occurs in a single vertical plunge; usually there are several rapids of greater or less extent between the dam or actual waterfall and the proposed site of the powerhouse, necessitating the construction of a canal or flume of greater or less extent, if the entire fall is to be utilized. Even with a waterfall in

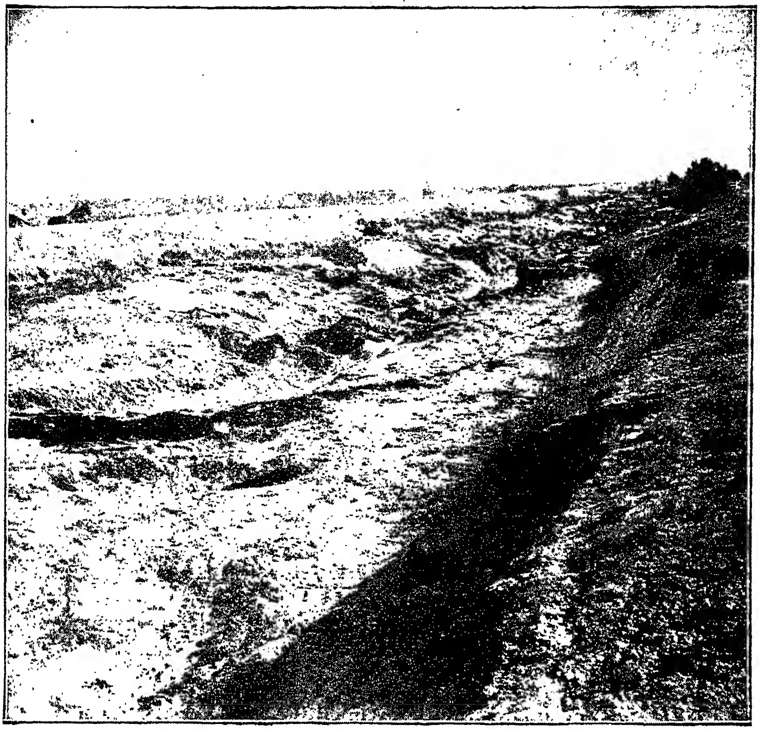


Fig. 165. Fowler Switch Canal, California, Showing Effect of High Velocities
Courtesy of U. S. Geological Survey

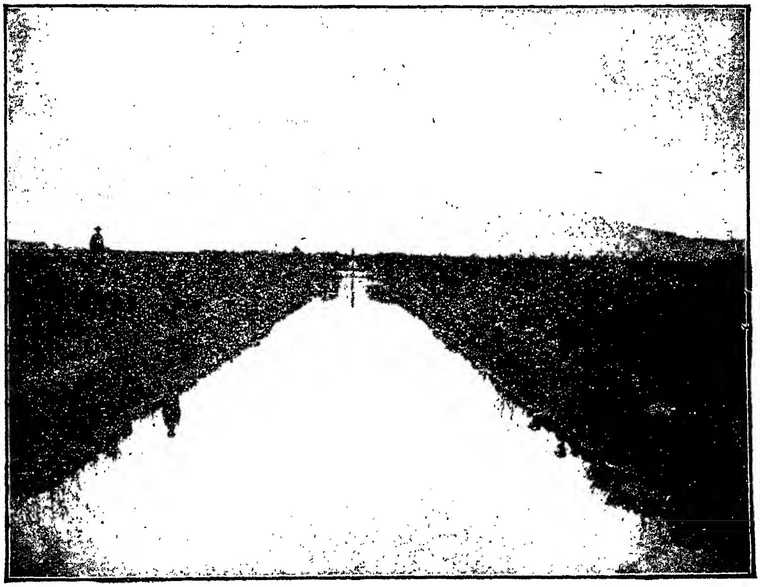


Fig. 166. Bear River Canal, Utah, Looking North
Courtesy of U. S. Geological Survey

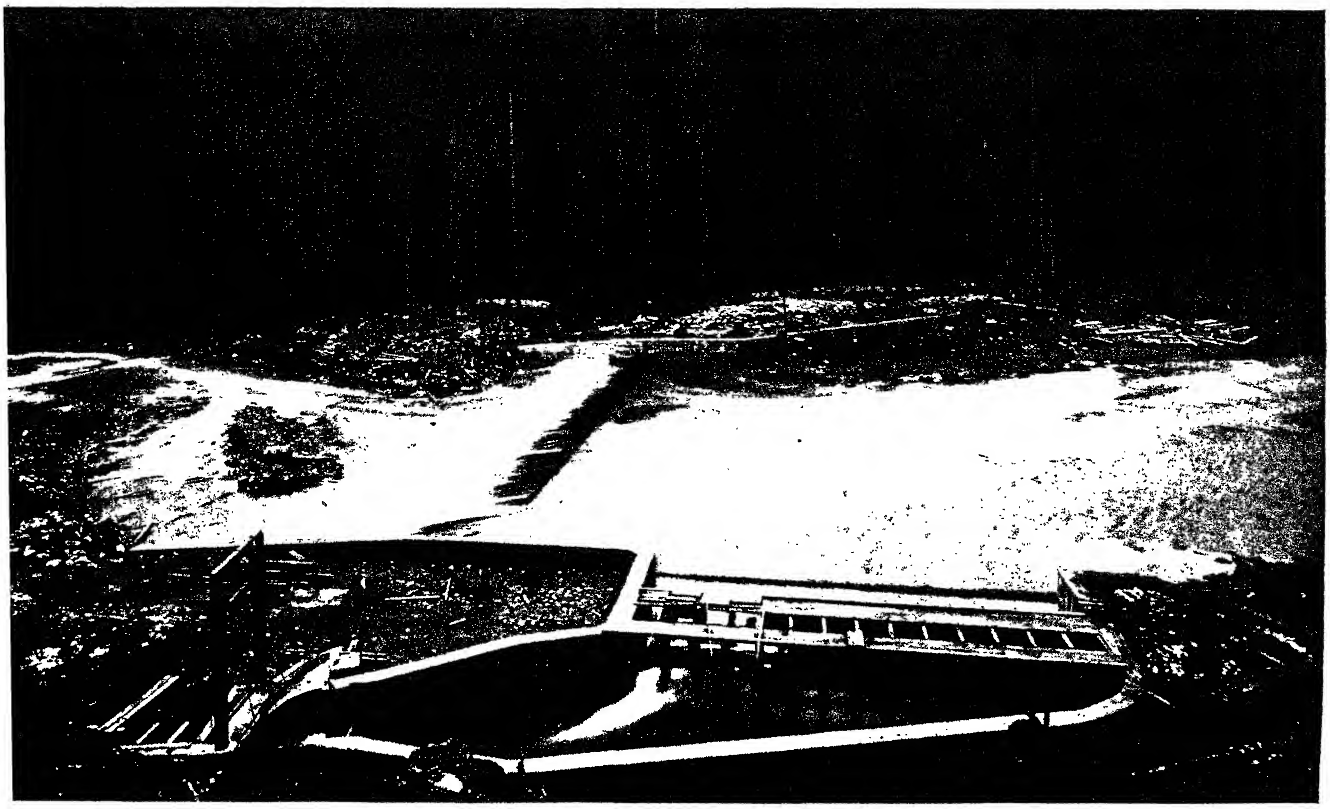


Fig. 167. Diverting Dam, Intake, and Head Gate for Flume
Courtesy of Pelton Water Wheel Company, San Francisco, California

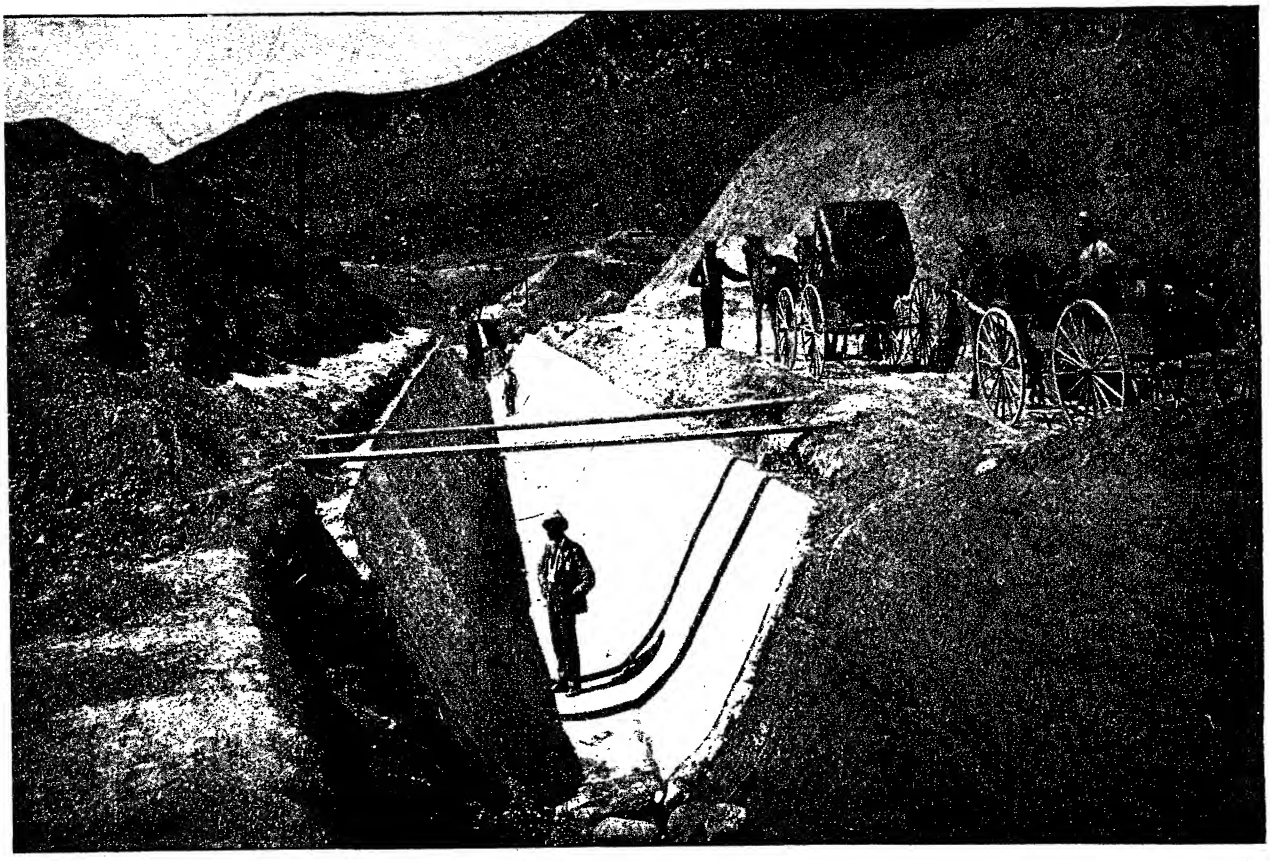


Fig. 168. Cement-Lined Santa Ana Canal, California; Capacity, 240 Second-Feet
Courtesy of U. S. Geological Survey

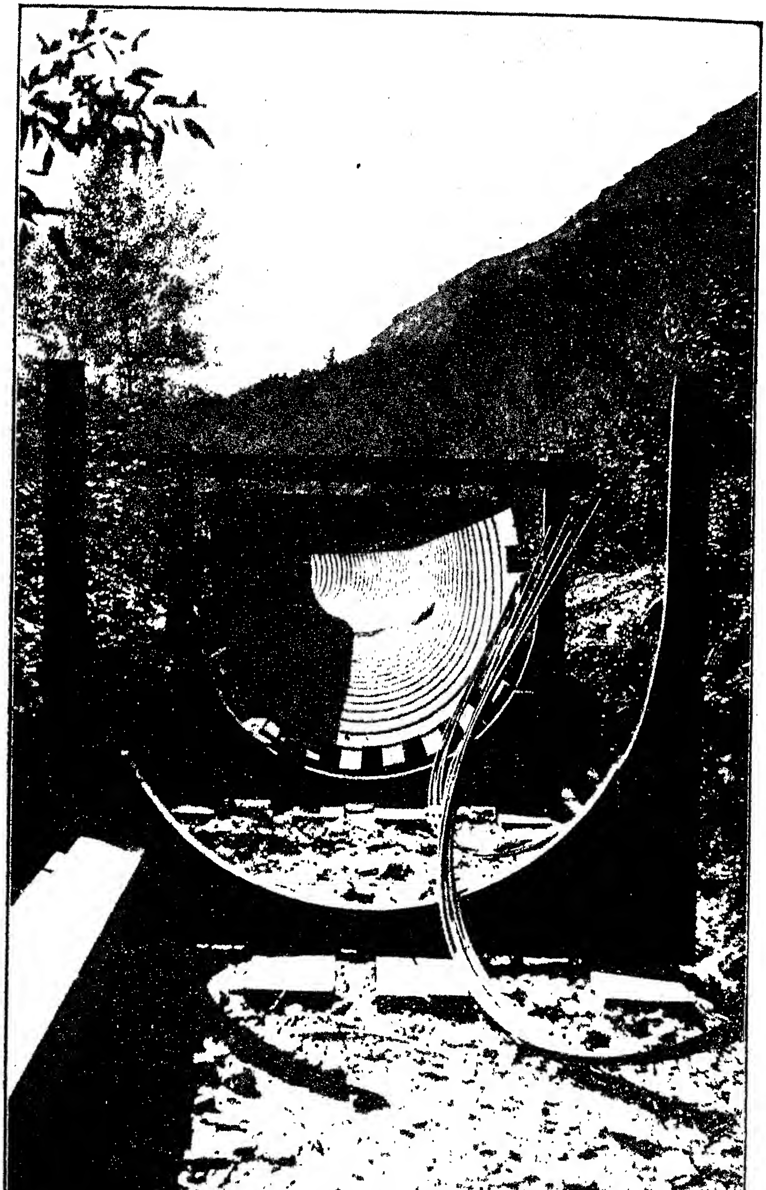


Fig. 169. End View of Sterling Flume in Provo Canyon, Utah
Courtesy of U. S. Geological Survey

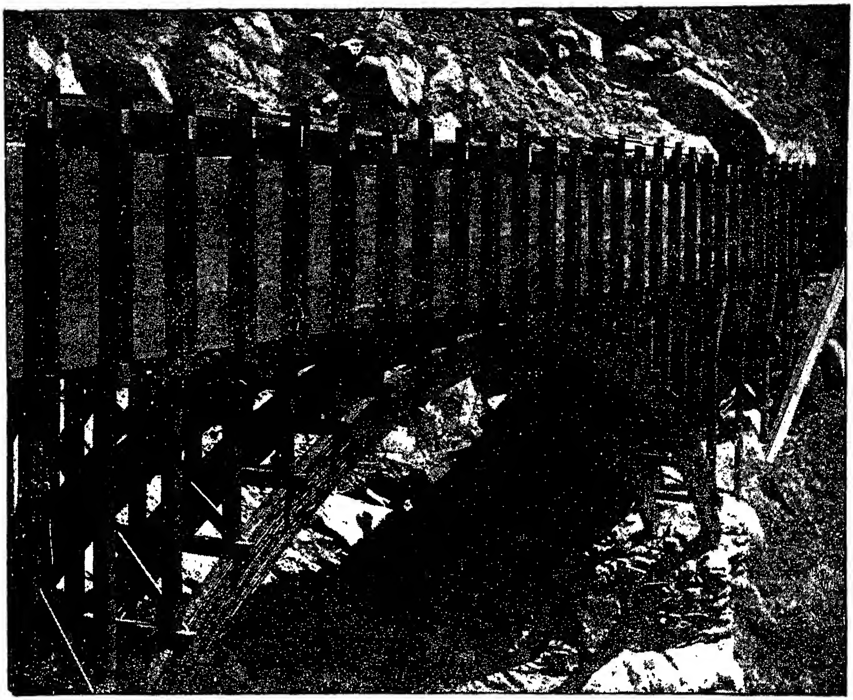


Fig. 170. Flume of Kern Valley Power Development Works, California
Courtesy of U. S. Geological Survey

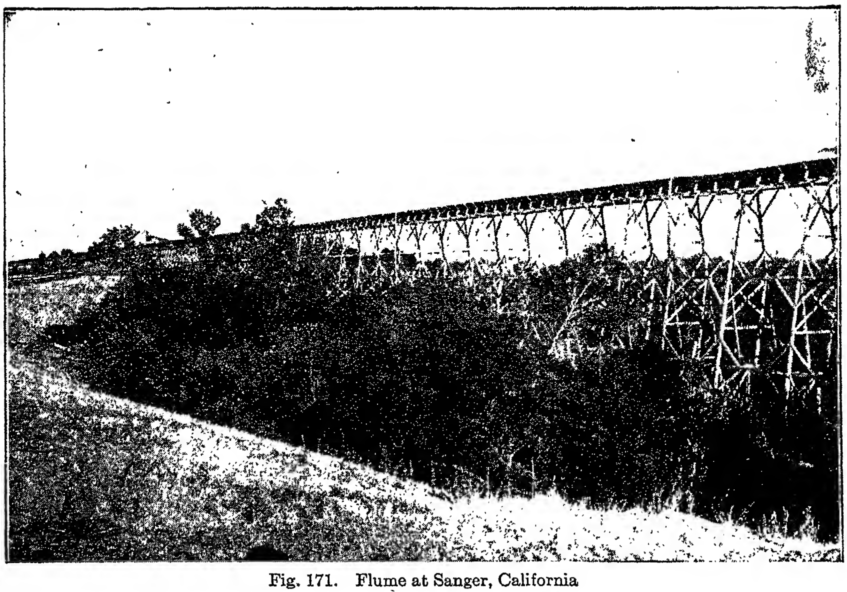


Fig. 171. Flume at Sanger, California
Courtesy of U. S. Geological Survey

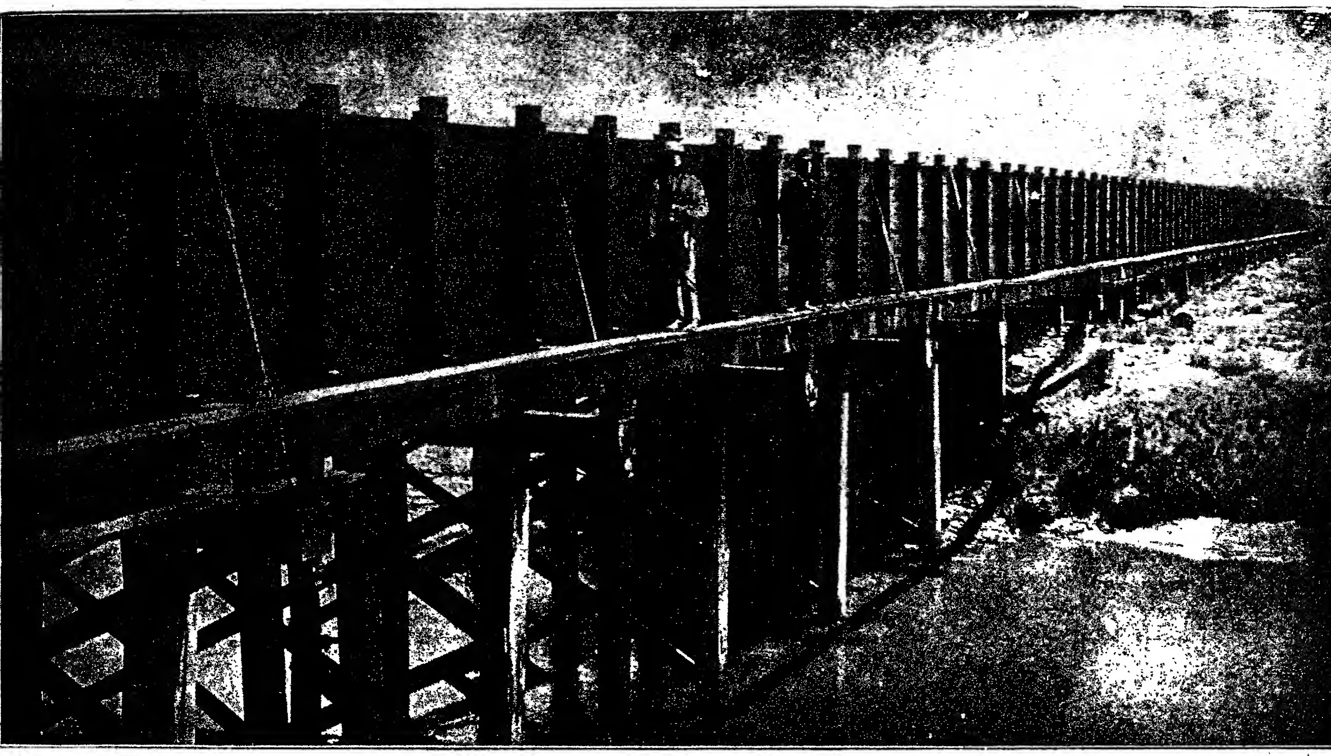
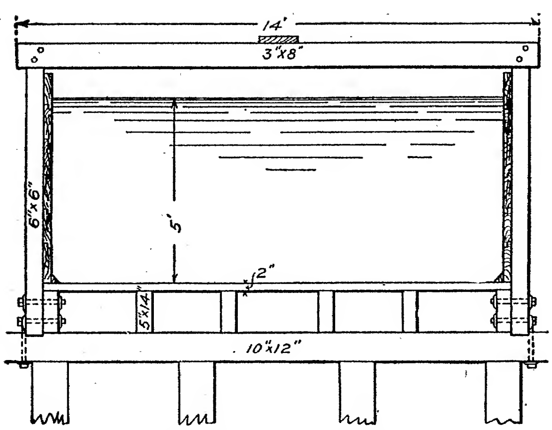
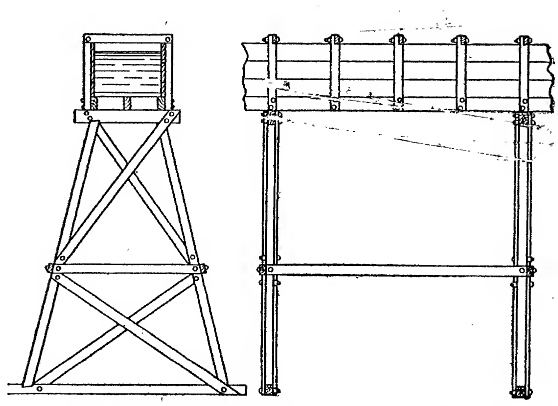


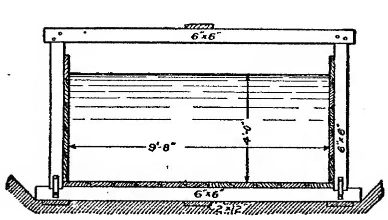
Fig. 172. Flume across Pecos River
Courtesy of U. S. Geological Survey



(A) Cross-section of flume resting on piles



(B) End and side elevations of flume on trestle



(C) Flume on steep hillside

ig. 173. Details of Flume Construction for Various Conditions of Support

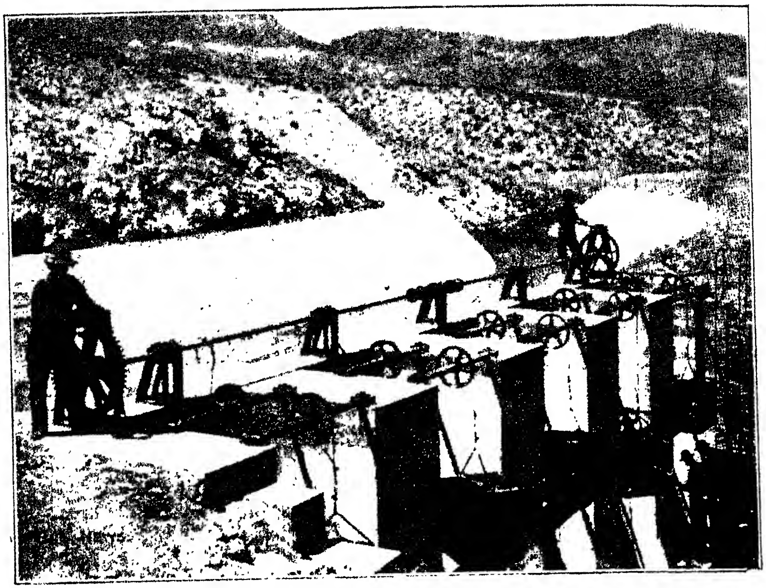


Fig. 174. Wasteway No. 1, Truckee Canal, Nevada, Showing Mechanism for Operating Taintor Gates

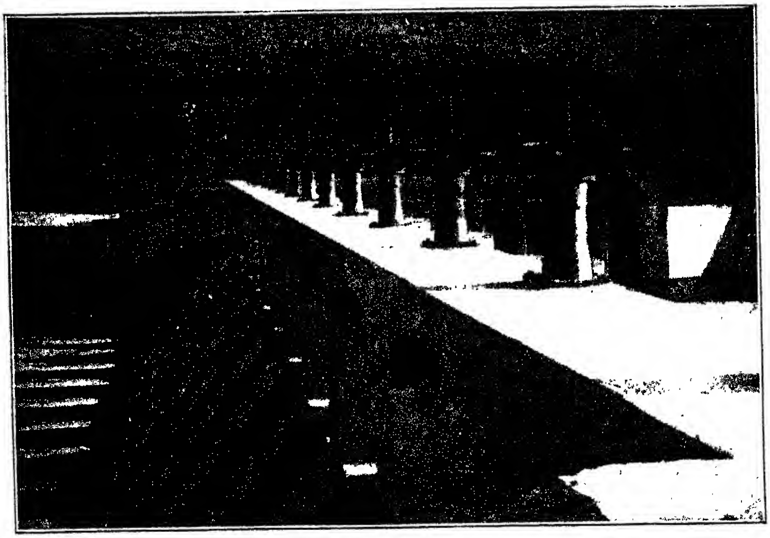


Fig. 175. Diversion Dam and Gates, Heading of Main Truckee Canal, Truckee-Carson Project, Nevada

View looking south along the dam, showing front face of dam; also the gate-operating mechanism.

which the total available head occurs in a single drop, a canal or flume is generally employed to convey the water to the several wheels or groups of wheels.

In a canal system, whether to be used for power purposes, water supply, or irrigation, there are many important features requiring very special attention, some of which have already been referred to—such as the headworks, with the corresponding regulator or headgates for controlling the supply of water into the canal head; the diversion dam or weir, of greatly varied construction, to raise the level of the water adjacent to the headworks and thus induce a proper flow into the canal; escape heads (wasteways) and their gates, to empty the canal quickly in case of accident or danger, or to dispose of surplus water in times of flood or excessive rains; sand gates, for scouring out deposits of sand or silt; vertical falls and inclined chutes, to compensate for excessive grade; and other features controlled by local conditions. Such features are treated in detail in connection with the subject of Irrigation.

The phenomena of erosion of bed and banks, of sedimentation of suspended matter, of capacity and velocity of flow with various cross-sections, and influence of kind of lining, together with their interrelations and mutual dependence, are discussed in connection with Hydraulics and Water Supply, and will not here be considered further than to present some typical illustrations; see Figs. 161 to 175, which, while not all associated with water-power developments, possess some features of general interest in this connection.

HYDRAULIC POWER INSTALLATIONS*

NIAGARA FALLS DEVELOPMENT

GENERAL CHARACTERISTICS

149. Great Lakes Resources. Along the boundary between Canada and the United States, there exists a chain of great lakes having a surface area of some 90,000 square miles, which receive the drainage from a catchment area of about 240,000 square miles. Between Lakes Superior and Huron there is a drop in elevation of the water surface of about 18 feet at the Sault Ste. Marie; between Lakes

*Proceedings of the Institute of Mechanical Engineers, Feb., 1906. Also *Engineering Record*, Jan., 1900; Nov., 1901; Nov. and Dec., 1903; Feb. and Oct., 1904; Apr., 1905; Oct., 1913; and *Electrical World*, Dec., 1910.

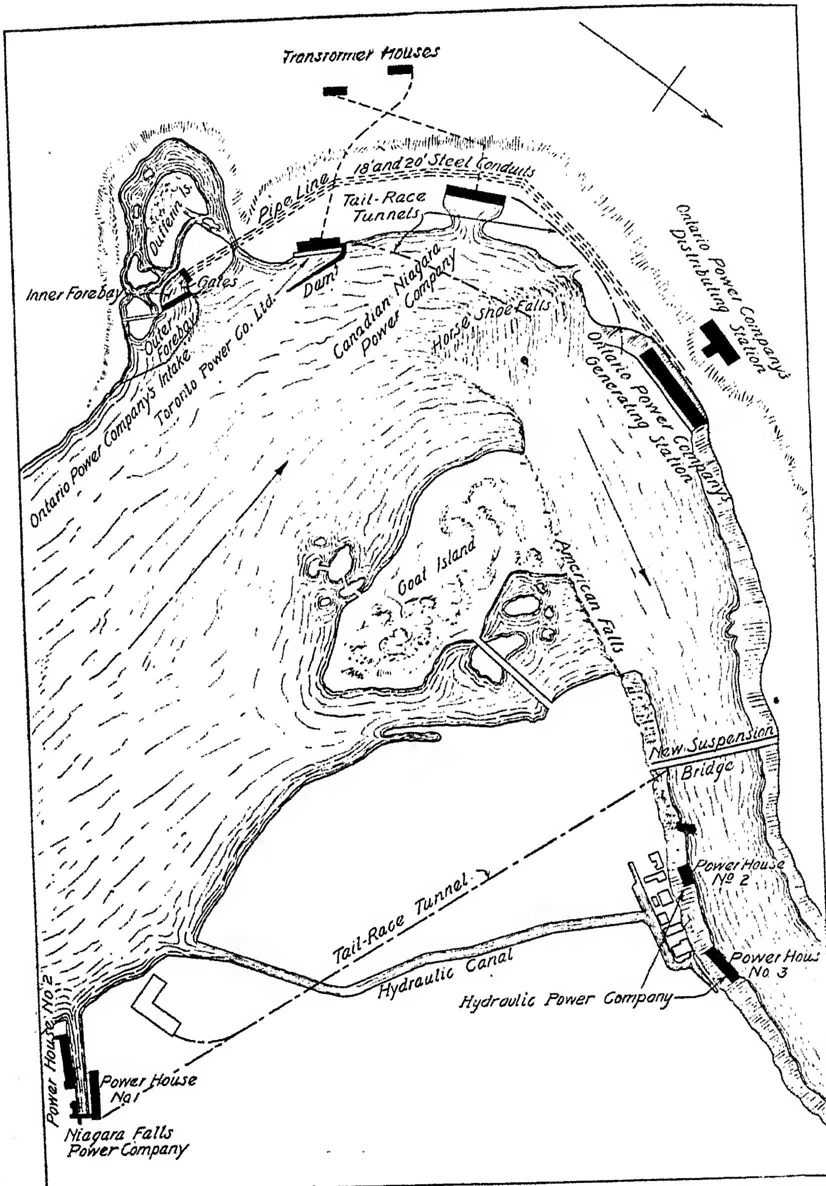


Fig. 176. Map of Niagara Falls and Vicinity, Showing Location of Power-Development Enterprises

Erie and Ontario, which are connected by the Niagara River, there is a total drop in elevation of 326 feet. The distance between these two lakes is about 30 miles, but almost the entire fall occurs in the last 15 miles. There is a fall of about 56 feet in the rapids above the Niagara Falls; about 160 feet at the falls; and about 110 feet below the falls. The entire drainage of the upper lakes flows from Lake Erie through Niagara River into Lake Ontario, and thence, by the St. Lawrence River, into the Atlantic Ocean.

These lakes form great natural storage reservoirs, so that the volume of flow and the levels in Niagara River are remarkably uniform. In extreme cases the river level above the falls varies $3\frac{1}{2}$ feet, the variation being chiefly due to wind holding back the outflow from the lakes. Below the falls, the river level varies at most 15 feet, due chiefly to ice blocks formed in the lower river.

150. Flow at Niagara. The minimum flow of Niagara River, as given by the government engineers, is 178,000 cubic feet per second; the mean flow is 250,000 cubic feet per second. The minimum flow, with the total fall of 326 feet, represents about 6,594,000 gross horsepower; with a fall of 216 feet (i.e., from the upper river to the foot of the falls), this would be about 4,369,000 gross horsepower. With the mean flow of 250,000 cubic feet per second, the corresponding figures would be about 9,261,000 and 6,136,000.

The falls comprise the Horseshoe Fall, about $\frac{1}{3}$ mile wide, on the Canadian side; and the American Fall, about 600 feet wide, on the opposite side, the two being separated by Goat Island. Below the falls, the river flows through a gorge or ravine 600 to 1200 feet wide, and 200 to 300 feet deep, eroded by the action of the river itself. Fig. 176 represents the conditions and locations of the various installations.

151. Early Development of Niagara Falls. The importance of the falls as a source of energy was recognized from an early period. The first important effort to obtain power was made in 1853, when construction on the so-called "Hydraulic Canal", 36 feet in width, 8 feet in depth, and 4400 feet in length, was begun from a point above the upper cataracts to a basin at the top of the bluff, located on the side below the falls. This canal was completed in 1861. On the bluff were constructed mills, having turbines supplied with water from the basin and discharging it through short tunnels on the face

of the bluff. In these cases, only part of the available fall (from 25 feet to 75 feet) was utilized, water being plentiful and the cost of excavating pits for the turbines considerable. In 1885, about 10,000 horsepower was utilized in this way, or the whole available supply of the hydraulic canal as then constructed.

HYDRAULIC POWER COMPANY

(Formerly Niagara Falls Power & Manufacturing Company)

152. Original Installation. In 1877 the hydraulic canal and all its appurtenances were purchased by the present owners under the name of the Niagara Falls Hydraulic Power & Manufacturing Company, the title being changed in 1910 to Hydraulic Power Company. In 1892 the company commenced an enlargement of its canal, and it has made notable improvements from time to time. The plan adopted at that time was to widen the original channel to 70 feet, and to make the new part 14 feet deep, thus providing an additional capacity of about 3000 cubic feet of water per second, giving a surplus power, after supplying the old leases, of about 40,000 horsepower.

Power House No. 1. In 1881 the first station, known as Power House No. 1, was established at this site for the purpose of supplying electricity for commercial purposes. It was located in the building later occupied by the Cliff Paper Mill. This station has since been abandoned.

153. First Section of Power House No. 2. In 1895–1896, a second power house was erected just below Power House No. 1, for the purpose of supplying power to customers. For this new plant a branch canal was taken to a forebay, 30 feet wide and 22 feet deep, near the edge of the bank. From this forebay, penstock pipes of flange steel, 8 feet in diameter, conduct the water down over the high bank a vertical distance of 210 feet, to the site of the power house on the sloping bank at the edge of the water in the lower river.

The first portion of the power house, 60 by 100 feet, was completed in 1896. Because of the fluctuations of the water in the lower river, it was necessary to place the floor of the station on which the generators stand, about 20 feet above the ordinary water level. As it was desired to couple the generators directly to the ends of the water-wheel shafts, it was necessary to place the water wheels also at

this elevation, and to employ draft tubes, in order to obtain the full head available. It was also required that the wheels should run at a given speed suited to the speed desired for the generators. To fulfil all these conditions, turbine wheels mounted on horizontal axes were adopted. The specifications for these wheels required that each should furnish 1900 horsepower, measured on the shaft of the wheel, when run at a speed of 300 revolutions per minute. The head under which the wheels work is generally 210 feet; but they were required to have sufficient capacity to deliver 1900 effective horsepower under a head of 205 feet; and all parts were to have sufficient strength to withstand the pressure due to a head of 220 feet without undue strain. They were required to show a percentage of useful

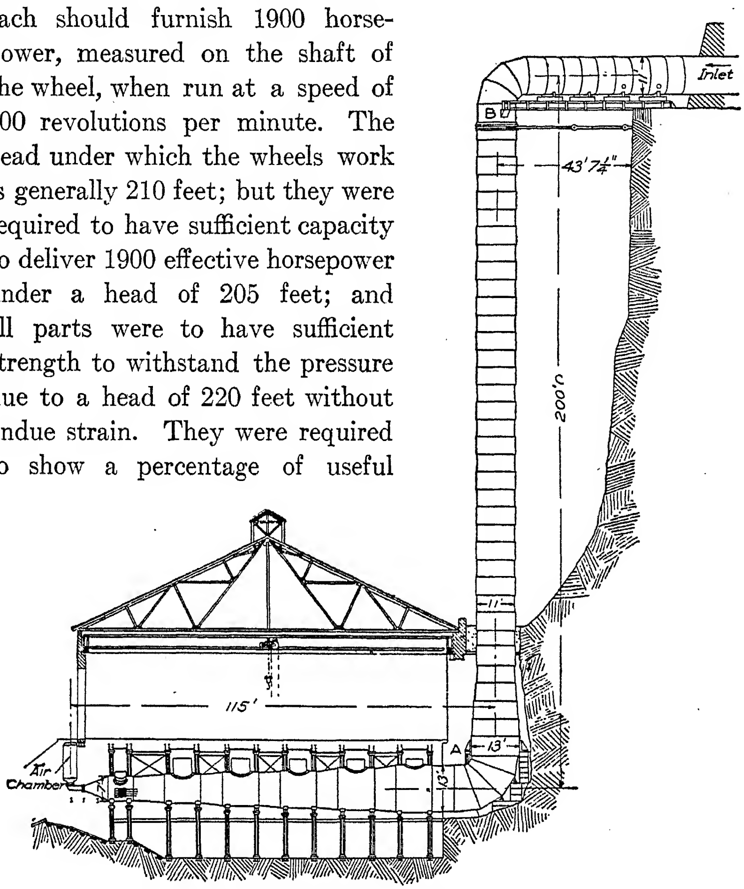


Fig. 177. General Dimensions of Penstock of the Hydraulic Power Company

effect of not less than 78 per cent, at any point between full and $\frac{3}{4}$ discharge, under any head from 205 to 225 feet and running at a constant speed of 300 revolutions per minute; and not less than 60 per cent under the same conditions, from $\frac{3}{4}$ to $\frac{1}{2}$ discharge.

The apparatus for regulating the speed of the wheels consists of a hydraulic piston which applies its force in either direction to a

rack connected with a pinion in the gate rigging of the turbine. The force which operates the hydraulic piston is air, compressed under about 15 atmospheres. This compressed air is contained in a cylinder, and the pressure is maintained by a pump which constitutes a part of the machine. The machine is provided with a high-speed ball governor actuating a balanced piston valve. The governor has an "anti-racing" appliance by which the governing machine is checked before it has carried the gate too far in either direction.

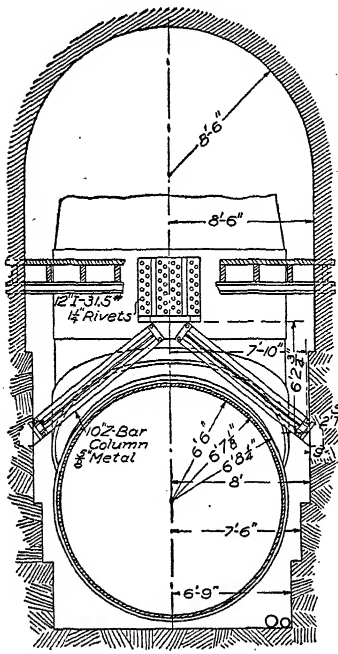


Fig. 178. Penstock Support in Plant of Hydraulic Power Company

154. Equipment of Second Section. The second section of this power station was completed in 1900, making the present size 120 by 100 feet. This portion of the station contains five turbines, each having a capacity of 2500 horsepower, Fig. 177. Power House No. 2, contains in all 16 turbine units, with an output of about 35,000 horsepower. The generators are of the direct- and alternating-current types. They are fed by a new 11-foot penstock consisting of a vertical portion about 200 feet high, with an arm on each end, that at the top having a length of about 68 feet, and that at the bottom about 115 feet. The penstock is built up in sections of 5 feet, the sections lapping inside and outside alternately. The thickness of the plates varies from $\frac{5}{16}$ inch at the top, to $1\frac{1}{8}$ inches at the bottom. Most of the sections are made up of the two plates. The thinner plates have lap joints with 2 rows of rivets; while the thicker plates have butt joints and double-cover splice plates with 3 rows of rivets on each side.

A vertical recess about 15 feet square and 50 feet high was cut out of the solid rock at the base of the cliff, and in it was set the bottom of the vertical portion of the penstock. At the lower elbow,

the penstock increases to 13 feet in diameter, and then decreases, as it passes the turbines, to a diameter of 7 feet at the outer end. The upper end has a bell-shaped inlet, 22 feet wide, built into the masonry of the forebay at an oblique angle. Heavy cast-iron brackets are riveted to the top of the vertical portion of the penstock on each side, and support one end of a pair of plate girders 30 inches deep and 49 feet long, which have 8-inch transverse I-beams across their top flanges to support the horizontal portion. About 6 feet below the bottom of the plate girders, the penstock is encircled by a pair of bent 10-inch I-beams with horizontal webs, between which plates are riveted to afford a pin connection for two sets of eyebar anchors which guy the penstock horizontally to 2-inch eye bolts drilled and cemented into the side of the cliff. At the lower elbow, the horizontal end of the pipe is seated on a bed of cut-stone masonry, and is bedded in cement mortar. The convex side rests on flanged cast-iron angle blocks riveted to the pipe and seated on stone piers. Large right-angled brackets of cast-iron, reinforced by heavy angles, are riveted around the pipe just above the elbow; and the one on the convex side rests on solid masonry, while the one on the concave side is supported by two inclined braces whose lower ends rest on castings let into the vertical rock wall.

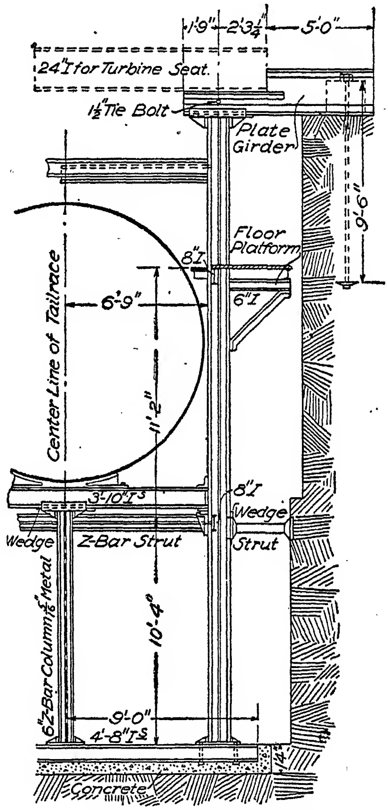


Fig. 179. Iron Work in Tailrace of Plant of Hydraulic Power Company

The tailrace is a rock cut 17 to 21 feet wide and 19 feet deep below the center line of the lower arm of the penstock, which is supported on steel columns. The lower horizontal part of the penstock is made

in sections which have alternate parallel and tapered sides. The former are uniformly about 10 feet long, and are fitted on top with vertical branches 5 feet in diameter. These branches are flanged to the inside of the penstock and riveted to it and to outside reinforcement collars, which are 8 inches wide and $2\frac{1}{8}$ to $3\frac{5}{8}$ inches thick.

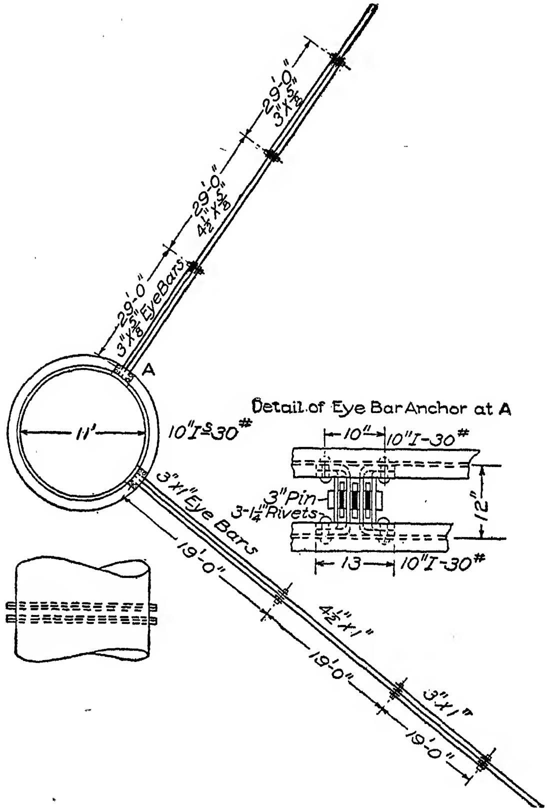


Fig. 180. Anchor at Top of Penstock in Plant of Hydraulic Power Company

A cross-section near the end of the penstock shows the knee-brace supports of the beam platform, the suspension rods by which the penstock is tied to the beams each side of the vertical branch, the anchorages of the double draft tubes to the masonry, and the transverse and longitudinal adjustable rods with which they are tied together just above high-water level, Figs. 178, 179, 180, and 181.

The valve gates are set horizontally, with their axes inclined 45 degrees to that of the penstock, and are operated by pneumatic pressure. The interposition of the tapered sections between the turbine connections reduces the diameter of the penstock in proportion to the diminished flow of water required as the successive turbines are passed, and brings it down to 7 feet at the last turbine, which is supplied through two side tubes. At this point the penstock is guyed laterally by four 5- by 1-inch horizontal eyebars on each side, which are anchored to steel bars drilled and cemented into the solid rock.

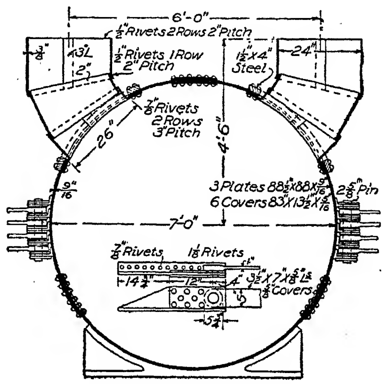


Fig. 181. Anchor Connections, End of Penstock in Plant of Hydraulic Power Company

The penstock terminates here with a conical section 7 feet long, which tapers to 18 inches, and connects by a cast-iron elbow with a vertical air chamber about 11 feet in extreme length, and 4 feet in internal diameter. It has an air valve on top, and gage glasses

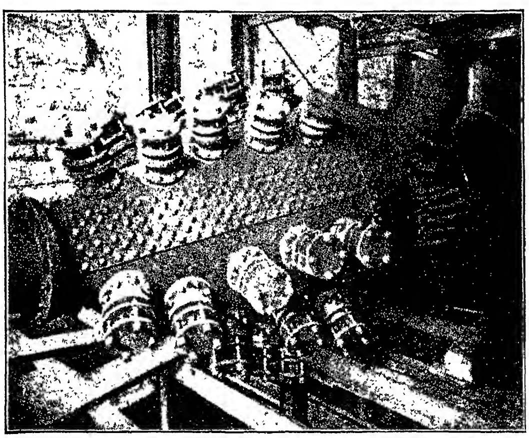


Fig. 182. Relief Valves on End of Penstock

on the side. The tapering portion is fitted with 30 spring relief valves, set in three groups, having 6-inch openings and set to open at a pressure of 100 pounds per square inch, Fig. 182.

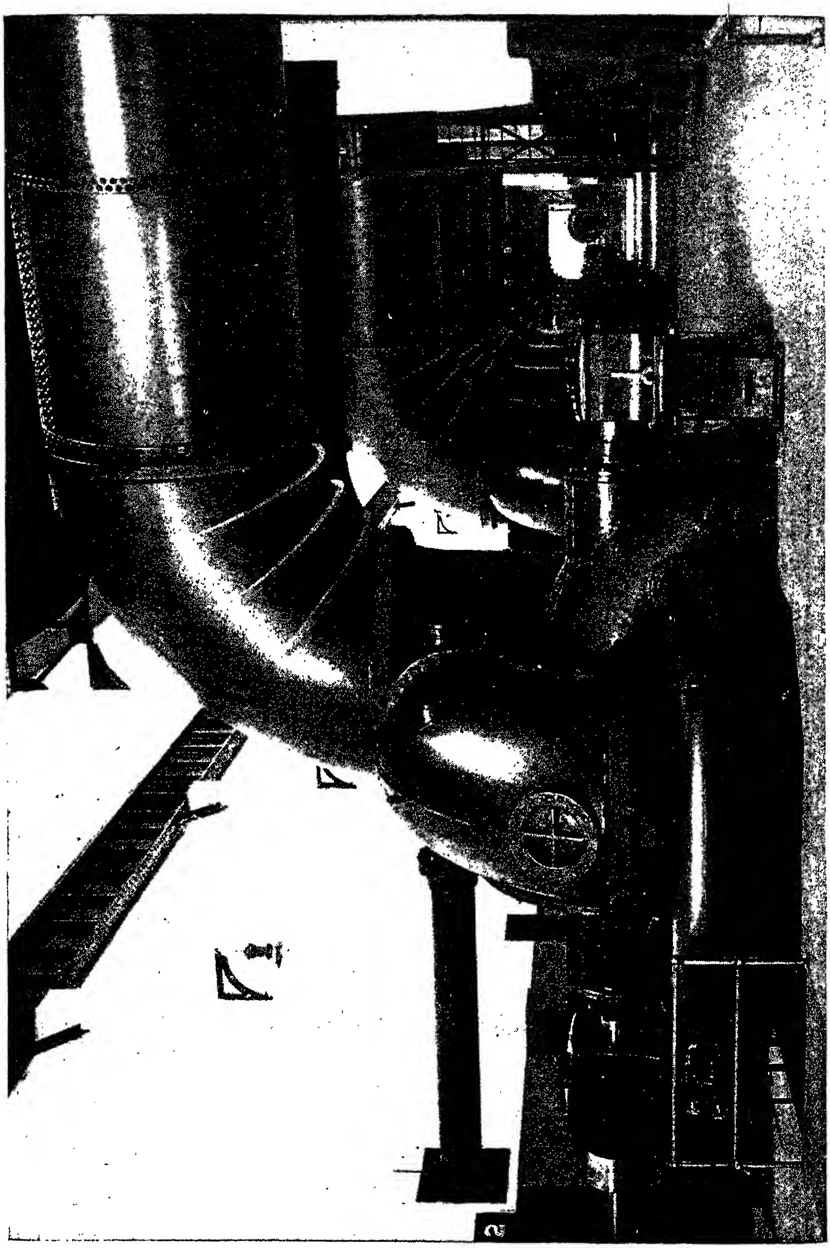


Fig. 183. View of Turbine Room, Station Number 3, Hydraulic Power Company, Niagara Falls, New York
Courtesy of I. P. Morris Company, Philadelphia, Pennsylvania

The turbine wheels are made of bronze, and are located in the draft-tube casing, one on each side of the casing proper; each pair weighs about 5000 pounds. From the sides of the turbines, the discharge pipes project laterally and then downward to connect with the draft tubes, which are 22 feet 8 inches long.

155. Power House No. 3. The third power house was begun several hundred feet below Power House No. 2 in 1905. A concrete wall separates the turbine and generator rooms. Each turbine (of the mixed-flow type) is supplied with water from the canal by a separate penstock entering the casing at the top, the water flowing around the entire circumference, passing inward and discharging horizontally to the right and left through draft tubes so arranged as to balance the horizontal thrust. The twin draft tubes of each turbine are also connected by pipes for the purpose of equalizing the horizontal pressure. The final installation consists of thirteen 10,000 horsepower turbines, Fig. 183, five of which are connected to 2 direct-current generators each, and eight to alternating-current generators. Two 1000-horsepower turbine-driven exciter units complete the present installation, which provides a total capacity of 130,000 horsepower.

NIAGARA FALLS POWER COMPANY

156. Power Installation. This was the first of the large electric-power developments at Niagara Falls, the general plan of development having been decided upon by an international commission of engineers. The engineering history of the power company began in 1889, when the Cataract Construction Company proposed to divert water from the upper river into an open canal at a point somewhat over a mile above the falls, to deliver the water to wheels in a pit at the side of the canal, and to conduct the water from the wheels to the river below the falls through a tunnel 7000 feet long, driven through the rock at a level nearly 200 feet beneath the city of Niagara Falls. This company began work in 1890, and first delivered power in 1895, with 3 units installed in Power House No. 1. The installation now comprises two power houses.

Power House No. 1. The first power house completed in 1900, had originally ten 5000-horsepower units, each unit consisting of twin Fourneyron inverted outward-flow reaction turbines without

draft tubes, with electric generators on the upper ends of the vertical shafts. These wheels were designed to give 5000 horsepower each when discharging 430 cubic feet of water per second, at 75.5 per cent efficiency, under a head of 136 feet, i.e., the distance from the surface of the headwater to the center line between the upper and lower wheels of the twin-turbine units, which discharged freely into the air above the tailwater in the tunnel, for at that time draft tubes did not seem applicable in this case. During the period 1910–1912 these ten wheels were replaced by an equal number of 5500-horsepower single Francis inward-flow turbines with draft tubes, similar to those which had already been installed and in use for several years in Power House No. 2, resulting in a 10 per cent increase in power capacity for the same quantity of water.

Power House No. 2. The additional power house completed in 1904, is, in its main features, similar to Power House No. 1, but contains 11 single Francis 5500-horsepower inward-flow turbines with draft tubes.

These wheels were designed to produce 5500 horsepower each under a head of 145 feet (utilizing the draft-tube head) when running at a speed of 250 revolutions per minute and using 445 cubic feet of water per second, this capacity in the turbines being intended to provide for a 10 per cent overload in a 5000-horsepower generator.

The aggregate capacity of these two power houses is 115,500 horsepower, and at present they utilize the total quantity of water granted to this company by the international treaty.

Reconstruction. The old turbines, the first of which was in commercial service in 1895, performed in a very satisfactory manner during their seventeen years of continuous service, and, upon dismantling, the runners were found in excellent condition, good, in fact, for several years longer. It, however, became apparent to the power company that by installing Francis-type turbines with draft tubes in place of the Fourneyron type the efficiency of the plant would be increased and its cost of maintenance decreased, both to an extent that would justify the change. The ten new units were installed and placed in service without interruption to the output of the plant, and well within the estimated cost and time of construction, and the increase in efficiency of the plant which was predicted has been fully realized.

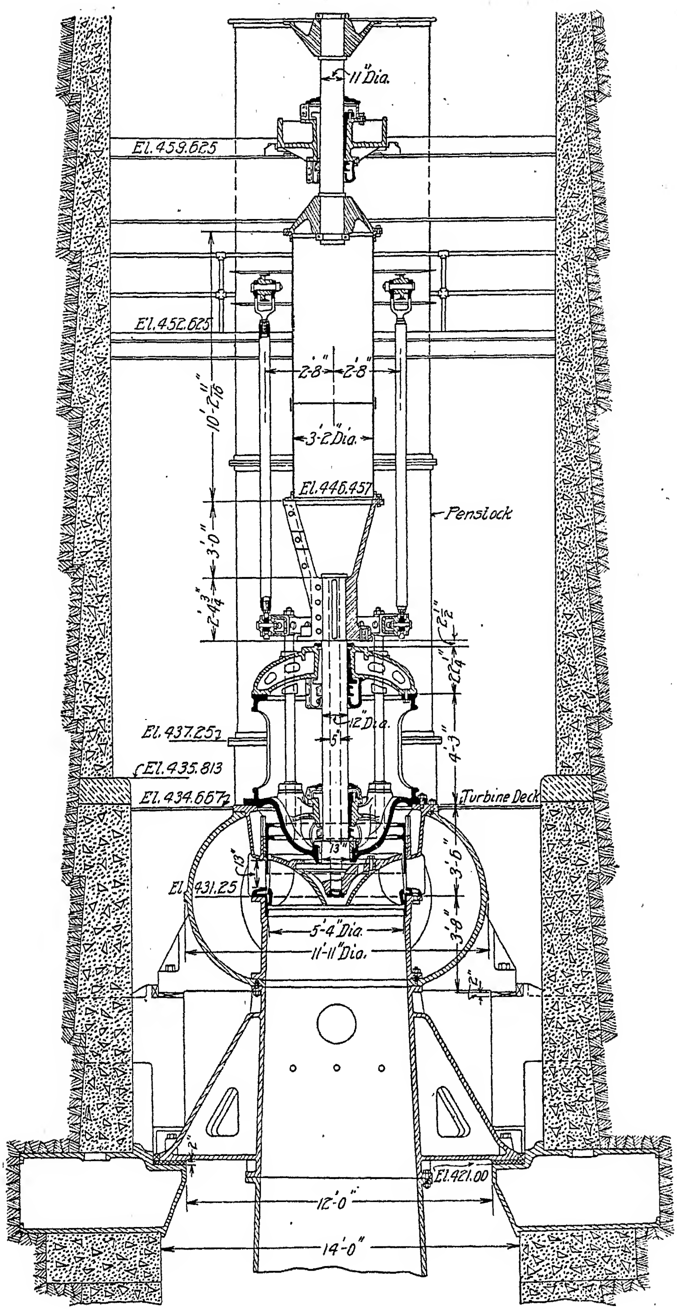


Fig. 184. Sectional Elevation of One of Ten 5500-H.P. Turbines for Niagara Falls Power Company
Courtesy of "Engineering Record"

The old penstocks which conduct the water to each turbine have been retained, although they had to be shortened to adapt them to the higher elevation of the turbines made possible by the use of draft tubes. The old shaft between generators and turbines, with its guide bearings, is continued in use. The thrust bearing, however, as well as the governor on each reconstructed unit, is new.

Each thrust bearing is of a combination oil-pressure roller bearing, which carries the entire revolving load of about 175,000 pounds at 250 revolutions per minute. Besides giving in reality two thrust bearings, each capable of carrying the load, it permits the use of a single oil pump to supply oil under pressure to both governor and thrust bearing, thus simplifying the high-pressure system. The governors are of the oil-pressure type.

Each reconstructed unit is provided with a triplex single-acting high-pressure oil pump, which provides oil for both thrust bearing and governor. Each pump is driven from the main turbine shaft through a rawhide gear and Reynold chain, thus eliminating practically all the noise usually found in power plants employing pumps driven from the main shaft.

The accompanying illustration Fig. 184, shows the arrangement adopted for units 1 to 4 in the wheel pit. Units 6 to 10 are similar in arrangement, except that the draft tubes are somewhat longer. The draft tubes of the turbines in the downstream end of the wheel pit discharge the water at a higher velocity than those in the upstream end, which results in a higher efficiency than if all the draft tubes discharged at the same velocity. The discharge of the draft tubes in the direction of flow in the wheelpit has greatly increased the efficiency of the plant over that with Fourneyron turbines.

157. General Construction. The canal is located on the New York shore about $1\frac{1}{4}$ miles above the crest of the American Fall, at a point where the mean water elevation is 561.2 feet above sea level. The discharge portal of the tunnel is about 1100 feet below the American Fall, at the mean elevation of 343.4 feet, thus giving a gross head of 217.8 feet. The canal, which was excavated to an average depth of 12 feet of water, and projected 600 feet beyond the original shore line into the river with embankments formed from the excavated material, has now a length of about 1500 feet, and a width of 180 feet at the mouth, and 100 feet at the inner end.

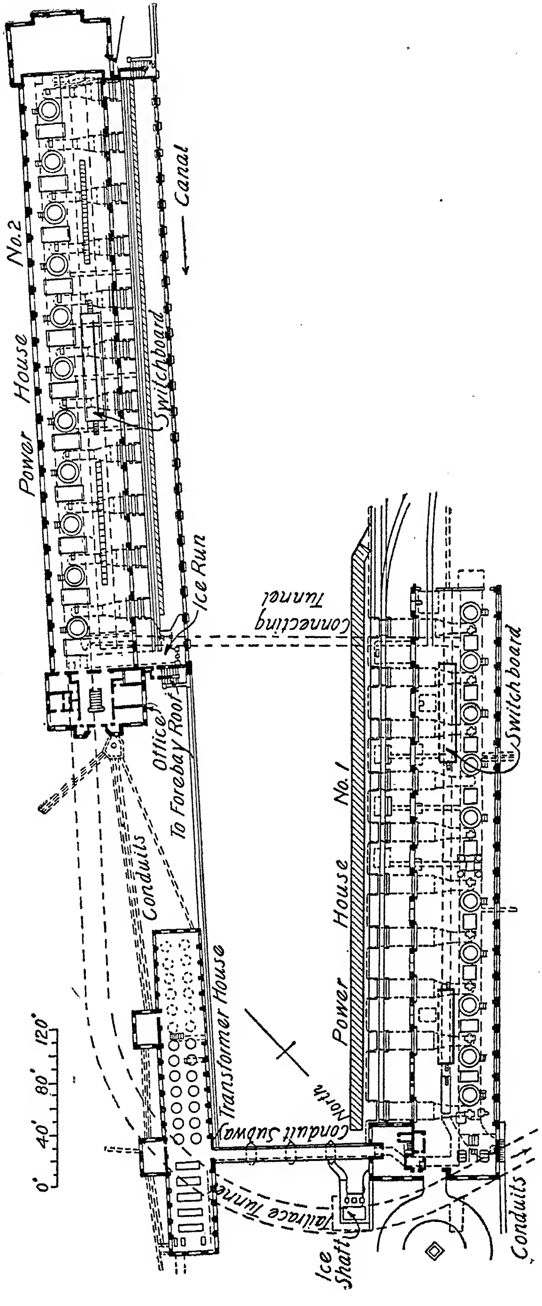


Fig. 185. Plan of Power Houses, Canal, and Upper End of Tailrace Tunnel of the Niagara Falls Power Company

The wheel pits—in which are located the penstocks conducting the water to the turbines, and the shafting connecting generators with turbines, and over which are built the power houses—are

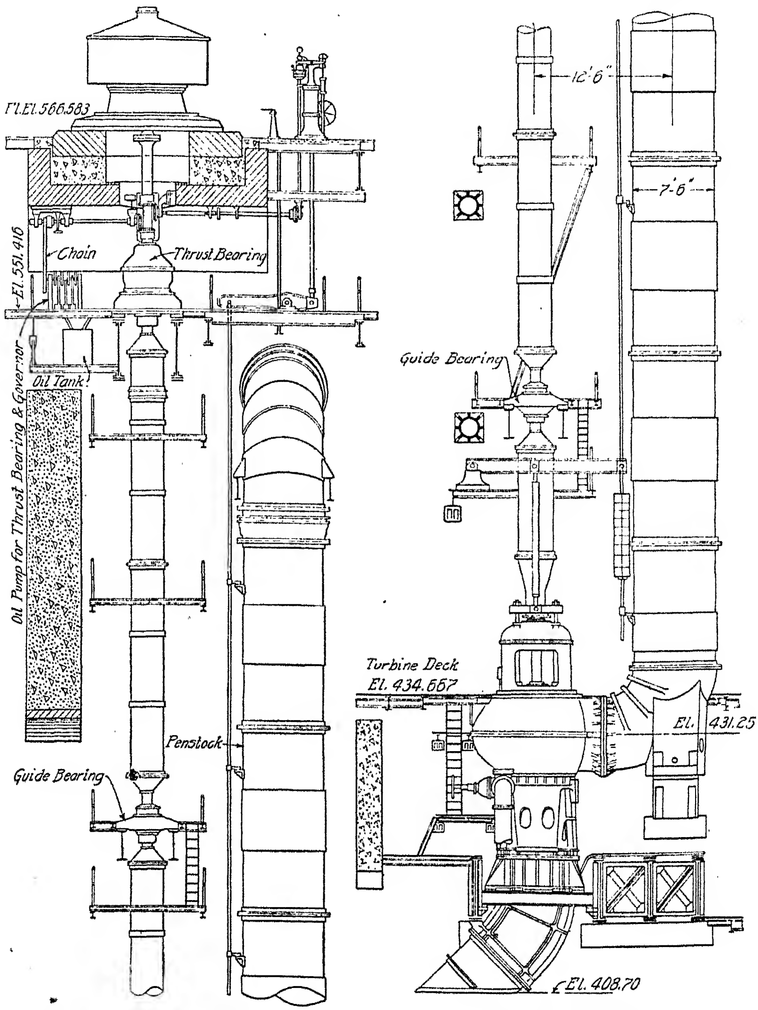


Fig. 186. Elevation of Turbine in the Pit at Power House Number 1, Niagara Falls Power Company
Courtesy of "Engineering Record"

located on opposite sides of the canal, as shown in the accompanying plan, Fig. 185. The pit for Power House No. 1 is 178 feet deep, 18 feet wide, and 425 feet long, and now houses 10 units of 5500

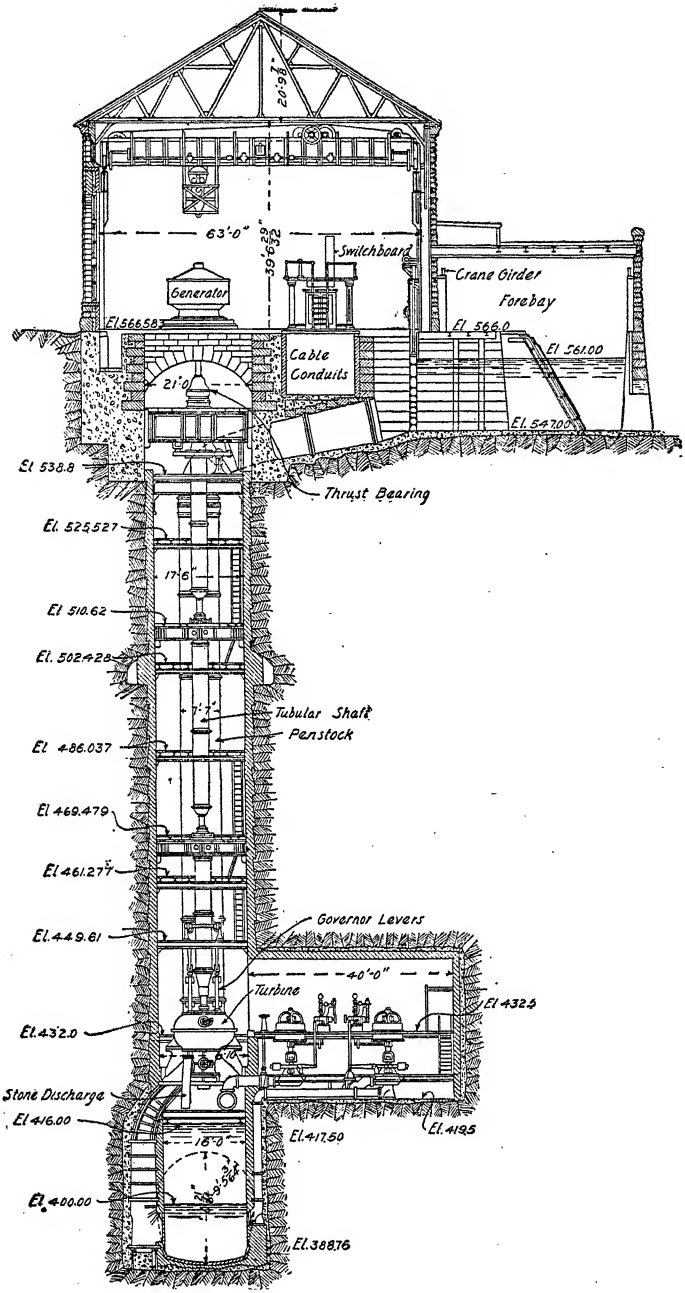


Fig. 187. Cross-Section of Power House No. 2 and Wheel Pit, Niagara Falls Power Company

horsepower, Figs. 184 and 186; the pit for Power House No. 2, Fig. 187, is 178.5 feet deep, 20 feet wide, and 468 feet long, and accommodates 11 units of 5500 horsepower each.

Each unit being supplied from a $7\frac{1}{2}$-foot penstock, the velocity in the penstock is approximately 10 feet per second. With 10 wheels in operation in each plant at one and the same time, the total quantity of water to be taken through them is over 10,000 cubic feet per second, equivalent to a velocity in the canal at its entrance of 4 feet per second.

The tailrace tunnel, through which all the water must pass, has a maximum height of 21 feet, and a width of 18 feet 10 inches, and a net cross-sectional area of 335 square feet. The mean velocity through it with the given quantity of water is, therefore, about 29 feet per second. The tunnel being about 7000 feet long, a considerable portion of the gross head must necessarily be employed in effecting the removal of the water at the required rate; it was accordingly given a slope which averages 6 feet in 1000; placing the floor of the tunnel at the wheel pit at the elevation of 387 feet, or 43.6 feet higher than the river level at the portal.

Canal and Forebays. The canal was excavated to allow a depth of 12 feet at low water, as already stated. The walls of the canal are of solid masonry 17 feet high, 8 feet thick at the base and 3 feet at the top, laid in an ordinary Portland-cement mortar composed of 1 part cement and 2 parts sand. Besides the two power houses which have to be supplied with water, the canal is also tapped for a supply to a separate wheel pit owned by the International Paper Company, whose property adjoins that of the Niagara Falls Power Company. Hydraulic power is sold in this case, as, at the inception of the work, electric power could not be furnished in time; and the power company provides for the disposal of the water through a 7-foot tunnel running from the wheel pit of the International Paper Company to the main tailrace tunnel.

The points of intake to the power houses, or outflow from the canal, are distributed in two groups to avoid local high currents as much as possible, the two power houses being, partly for this reason, located diagonally across the canal. There is a separate opening for each penstock in each of these power houses, but the intakes are materially different. Racks of the usual flat iron-bar construction

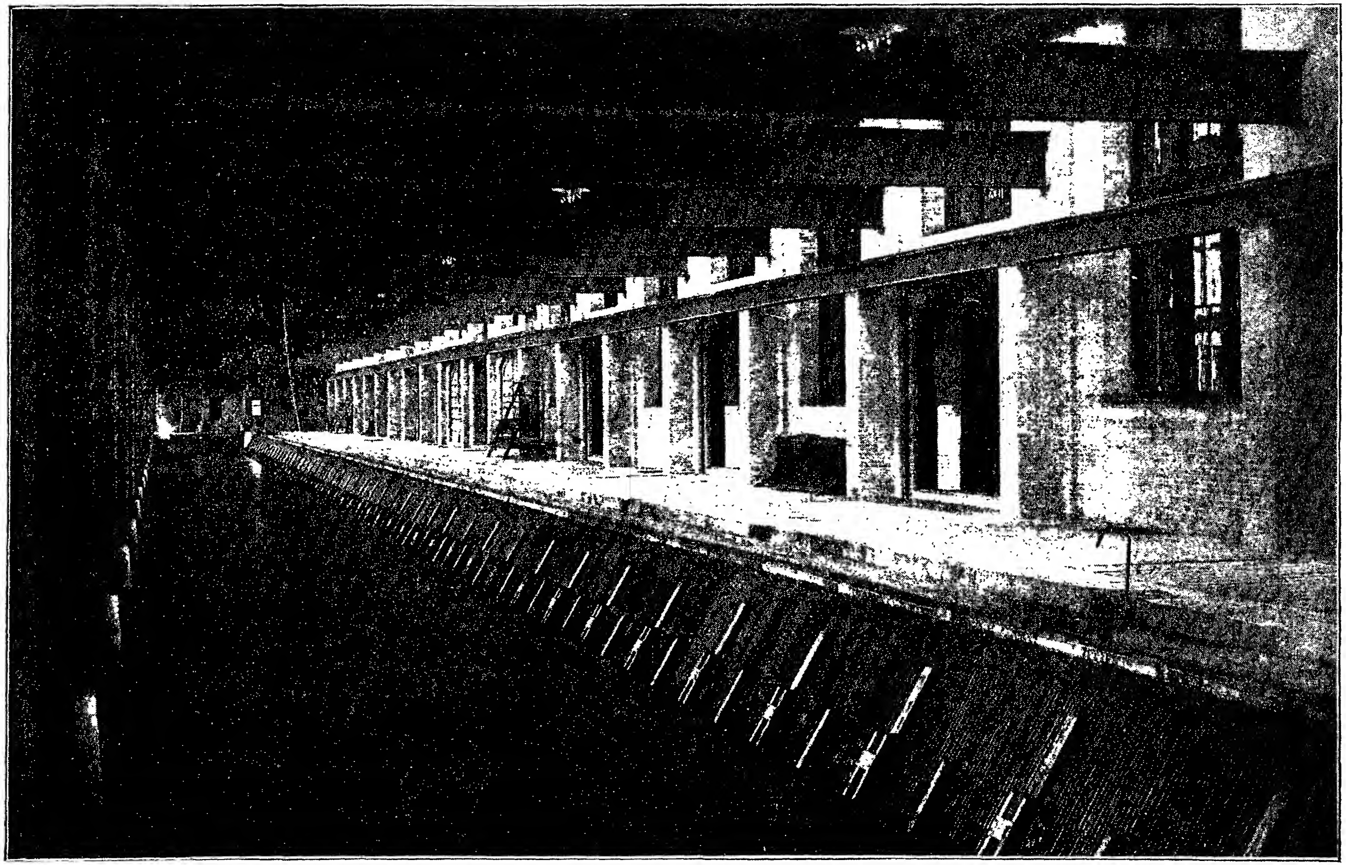


Fig. 188. Interior of Forebay of Power House No. 2, Niagara Falls Power Company, Niagara Falls, New York
Racks are shown protecting entrance to turbine chambers

guard the entrances to the penstocks; but in the later installation they are enclosed by a portion of the power house, in a covered forebay, Fig. 188. While the extra cost of a covered forebay is considerable, its provision was deemed advisable as a means of fighting ice. Under ordinary circumstances, there is a floating timber boom extending across the mouth of the canal; but cakes of ice, especially with a large flow of water, find their way under the boom and into the canal. During the present year the boom has been located a considerable distance in front of the canal, to divert ice more readily toward the falls.

Much trouble used to result from allowing floating ice to enter the canal, and a large corps of men was constantly required in cold weather, several shifts per day, cleaning the racks at Power House No. 1. These racks consisted of three parts: a bottom section; a top section, dropping from the top of the canal wall to a level about 1 foot below water; and a middle removable section entirely submerged, which could be temporarily hauled out of position for purposes of cleaning, etc.

Though a secondary boom was provided, extending into the canal a few feet in front of the rack and across the entire series of openings, cakes of ice would succeed in getting under this boom, and also would be drawn frequently under the top section of the racks, owing to the suction of the inflowing water. As the original wheels of the first power house were not designed to allow for an accumulation of ice, as will be explained, it can readily be seen that the passage of much ice into any penstock was likely to be a serious affair.

It was these general considerations that affected the design of the wheels of the second power house, and the arrangement of the intake at that building. Here the secondary boom is replaced by the outside wall of the covered forebay, the water being admitted into the forebay through arched openings, the crowns of the arches being about 4 feet below normal water level, this distance being assumed great enough to prevent blocks of ice driving through the arches.

In the design of the new power house, moreover, an ice run has been provided; and blocks of ice collecting in the canal are directed toward the run, and drawn by the current of water in the ice run down a shaft and into the tailrace tunnel. In a similar way, there is now

an ice run for Power House No. 1, secured by utilizing a shaft that had been sunk in connection with the driving of the extension of the tunnel to the second power house. It will thus be seen that much has been done to overcome the ice troubles, and that in this connection the tailrace tunnel has to take care of the additional amount of water, if it is necessary to prevent any great lodgment of ice in the intake canal.

Intake of Power House No. 2. The arches through which the water enters the forebay of the new plant from the canal are sprung from 5- by 6-foot piers, spaced 20 feet apart, and leaving openings 14 feet wide and 10 feet to the crown of the arch. This gives an area of flow of about 125 square feet; and as there are 2 arched openings per unit, the inlet area is 5.7 times that of the penstock, not allowing for the water taken from the forebay for the exciter turbines and other purposes.

Immediately inside the arches are grooves on each side for stop logs for shutting out water from the forebay at any time. A short distance beyond these, the water passes through the racks, which are of the usual construction, inclined 30 degrees with the vertical. The rack bars are of mild steel, 3 by $\frac{3}{8}$ inch in size, separated by pieces of $\frac{3}{4}$-inch gaspipe into spaces $1\frac{13}{16}$ inches wide. They are built in three sections as regards height—the upper, with rounded top, being about 8 feet long and extending 3 feet below normal water level; the middle, 10 feet long; and the bottom 2 feet 10 inches long. The center sections are removable, sliding between I-beams and channels; and for handling them, there is a 5-ton Niles electric crane traveling the entire length of the forebay. The rack structure is supported on 15-inch I-beams on 8-foot centers, and the bars are divided into two groups between them. The total wetted area of openings between the rack bars is substantially equal to that of the aggregate area of the arched openings. Between the racks and the head gates, which are in the power house proper, are two sets of grooves in the buttresses against which the racks are supported. These allow for closing the inlet to the corresponding penstock with stop planks, in the event of a need for repairs to the gate.

Gates. The head gates in the new power house, like those in Power House No. 1, may be operated both by hand and by electric motor. They are lifted by screws; but the new gates are provided

with wicket gates for passing the water into the penstock and relieving pressure before opening, and are not provided with the roller bearings used instead in the old plant.

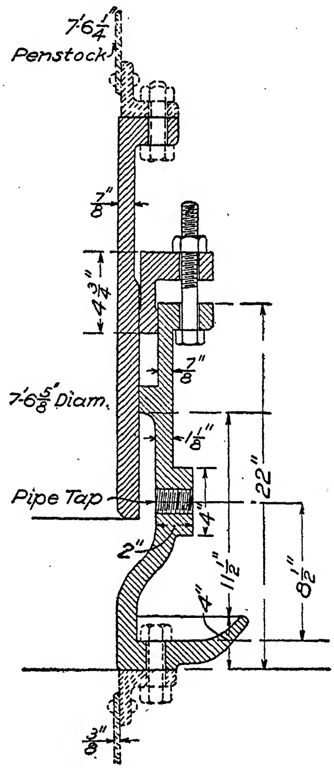
The ice run from the canal at Power House No. 2 is also provided with gates, so that no water may be wasted in this way during summer, as it is undesirable to have more water than necessary discharged into the tailrace tunnel. The ice run has direct connection with the canal, and also with the covered forebay, the latter inlet to the ice run being provided to take care of any ice which may succeed in entering the forebay. These gates are also designed to be lifted either by motor or by hand. They are of the lifting-screw type, and the motor drives a horizontal shaft which carries at each end a pair of bevel gears. The horizontal gear in each case revolves on ball bearings, and the screw is lifted and lowered through it.

Penstocks and Wheel Pits. The intake proper to each pen stock—that guarded by the head gate—is 14 feet wide, and under normal conditions carries 14 feet of water. The mouthpiece to the penstock starts a few feet back of the gate, with a flaring elliptical entrance, and, pitching 2 inches in a foot, joins the 90-inch circular penstock 14 feet beyond. The elliptical entrance is 12½ feet wide and 8½ feet high. The inclined portion of the penstock and the mouthpiece are bedded in concrete, and are arched over by brickwork which supports the substructure of the power house proper. The junction between the walls of the chamber behind the gate and the penstock mouth is made with cement mortar, and the joint between the inside masonry wall of the chamber and the brick wall of the cable conduit behind is waterproofed with asphalt.

From the mouthpiece to the turbine, the penstock comprises a short length of straight riveted-steel pipe; a riveted-steel elbow at the top of the wheel pit; six vertical sections of straight pipe, each 15 feet 5½ inches long and 90 inches inside diameter; and a cast-iron elbow at the bottom connecting into the turbine. The upper elbow is provided with cast-iron brackets, and the portion of the penstock above it is supported from it by the brackets, which bear on steel beams bridging the wheelpit. The rest of the penstock is carried by the cast-iron elbow at the bottom. There is a stuffing box of cast iron in the pipe immediately below the upper elbow, to allow for expansion; it consists of two rings of cast iron bolted to the abutting

ends of the penstock, one sliding on the other, as shown in an accompanying detail, Fig. 189. The stuffing box was required to stand a hydraulic test of 25 pounds. An idea of the size of the bottom supporting elbow of the penstock can be obtained from the fact that the metal is $2\frac{1}{4}$ inches thick, and it is reinforced by circumferential ribs 4 by 4 inches in size.

The steel of the penstock is $\frac{3}{8}$ inch thick in the upper part, and $\frac{1}{2}$ inch in the lower; and each section has two 18-inch manholes. There is a series of floors or decks and intermediate platforms in the wheel pit, as indicated in the drawings, and the penstocks and shafts are thus accessible throughout their height. The pit is lined with brick throughout, and cast-iron brackets are laid in the walls to support the various beams. Drainage pipes emptying into the tailrace were provided behind the wall, laid in broken stone and wherever ground water was likely to collect.



No. 189. Penstock Expansion Joint, Niagara Falls Power Company

Tailrace Tunnel. The tailrace tunnel, from a point near the wheel pit of Power House No. 1 to the portal at the lower river, was built in a straight line for construction and hydraulic reasons. Surveys for this purpose, which were started in the latter part of March, 1890, marked the beginning of actual work on the Niagara Falls Power Company's plant. The work was executed from the discharge portal, and from two shafts sunk at points 2600 and 5200 feet from the portal.

The tunnel is lined throughout with brickwork, partly on account of the poor character of the rock, and also because of the decreased friction to the flow of water thereby secured. The invert was laid last, and has a face of vitrified paving brick. The sides and arched top are usually four rings of brick 16 inches in thickness, but are

sometimes 6 and 8 rings thick; and the space behind was filled with rubble masonry. The bricks were laid in a mortar of 1 part Portland cement to 3 of sand, except where the flow of water is very great, in which case the proportions are 1:2, and in some cases 1:1.

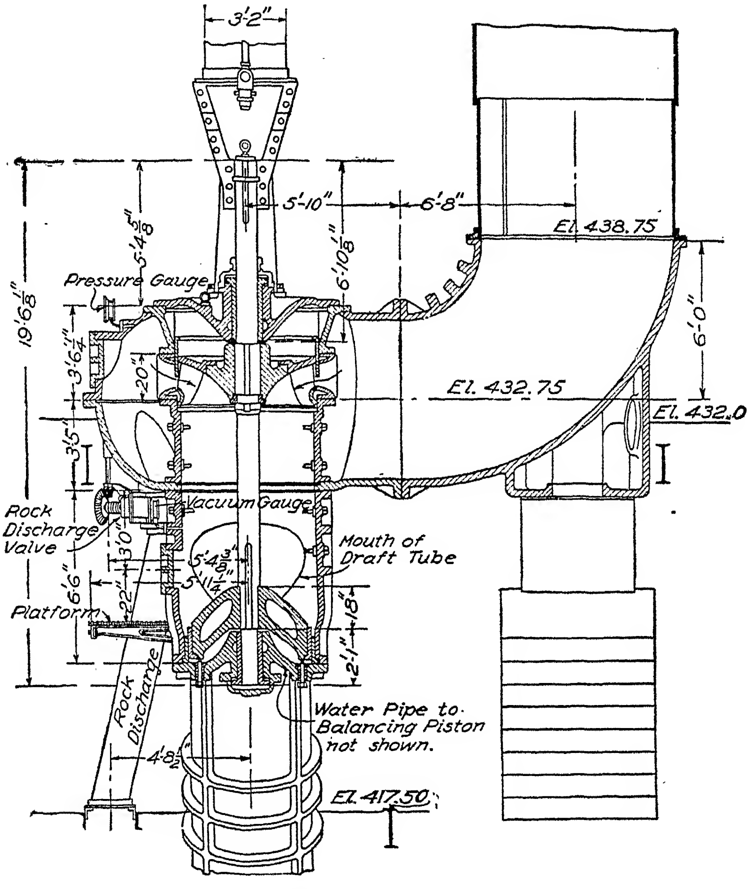


Fig. 190. Detail of 5500-H.P. Turbine in Power House No. 2, Niagara Falls Power Company

One of the most interesting points in connection with the tunnel is the provision of an ogee discharge. It provides for lowering the grade of the invert about 11 feet below the average low water of the river, so as to allow about one-half of the flow from the tunnel to discharge below the surface. The ogee surface starts at a point 95 feet from the portal, and the 10½-foot drop takes place in this dis-

tance. This portion of the tunnel, to the elevation of the spring line, is lined with steel boiler plate riveted to steel ribs 3 feet to 4 feet in depth, which are bedded in Portland-cement concrete. For the last 25 feet of the tunnel, granite masonry is used instead of the brickwork, and the arch and the face of the portal are also of

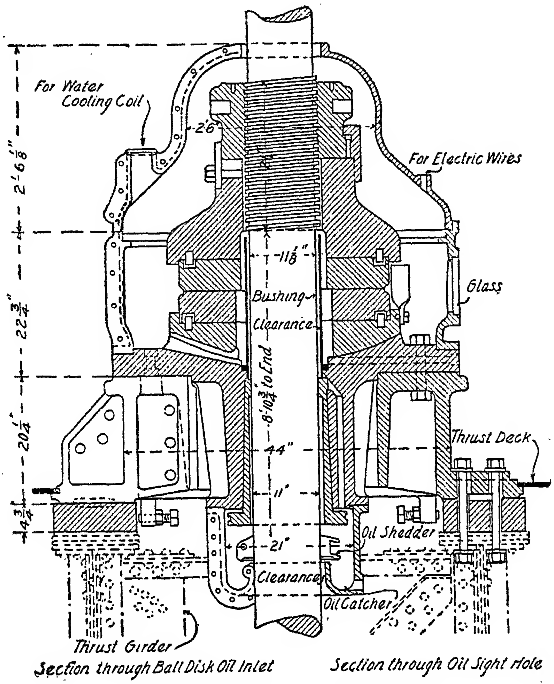


Fig. 191. Thrust, or Hanging Bearing

granite masonry, carried to 38 feet below the water surface to a ledge of white sandstone.

158. Turbines. A feature of the new turbine is the enlarged casing around the guide blades, which allows for collecting stones that may be carried down the penstock, and from which they may be discharged into the tailrace as desired, through a pipe dropping from the bottom, Fig. 190.

The manipulation of turbine gates is similar in the two plants, in that the governors are connected to the gates by a series of levers and suspender rods extending from the governors in the power house to the gates at the turbines.

The turbine shaft in the new plant, like that in the old, is built up of sections of pipe 3 feet 2 inches in diameter, with walls $\frac{3}{8}$ inch thick, connected by solid pieces of shaft 11 inches in diameter at the points where bearings are employed. Besides the two bearings at the turbine, and those at the generator at the top of the shaft, there are three guide bearings, and the thrust or hanging bearing, Fig. 191. The sections of tubular shaft are fixed to the solid pieces by hollow steel cones bolted around the 11-inch shaft, and to a ring of angle iron riveted to the tubular shaft. The guide bearings are located roughly 40 feet apart, and are 20 inches long. They are lubricated with oil under slight pressure; and a feature of their design is the use of oil shedders and oil catchers, to prevent oil from dripping downward along the shaft.

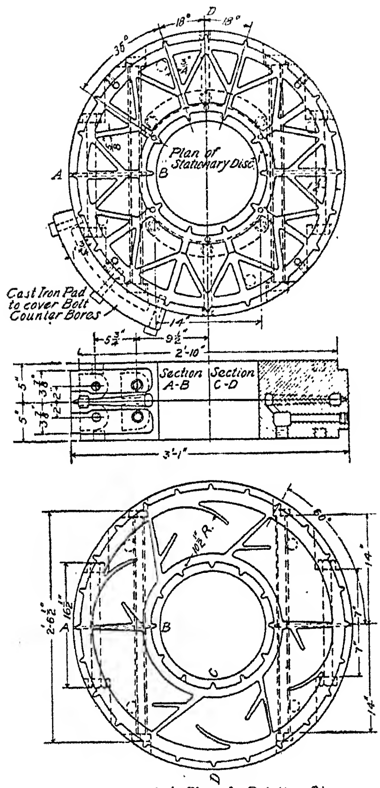


Fig. 192. Diagram Showing Oil Tubes in Thrust-Bearing Disc

Thrust Bearings. The choice of a thrust bearing has been the subject of a considerable amount of experimental work largely carried on at the power house. The plans called for a disk type of bearing, in which the shaft was hung by a revolving disk carried on a stationary disk. Three different types were constructed and placed on the shafts of the first three units of Power House No. 2. The disks were all made of close-grained charcoal iron, of 25,000 pounds tensile strength, and with the mating faces scraped to a bearing. Their differences arise from the method of lubrication. For wheel No. 11, the first in the new plant, forced lubrication was adopted; for No. 12, a combination of forced and self-lubrication; and for No. 13, self-lubrication alone.

In the combination bearing, arrangements are made to utilize two different pressures, the low-pressure oil to come from a general lubricating system, and the high-pressure from an individual pump on the thrust deck, of 25 gallons capacity per minute at 400 pounds pressure. The oil is introduced into the stationary disk at two diametrically opposite points, and the oil forced between the surfaces. Each disk has two circumferential grooves, one near the outer, and the other near the inner edge. Connecting these in the stationary disk, are grooves $\frac{3}{8}$ inch deep, and branching grooves $\frac{5}{16}$ inch deep. The connecting grooves of the rotating disk are bent backward and are $\frac{1}{2}$ inch deep near the inner ring, Fig. 192.

The thrust disks of the bearings in all cases are enclosed in a casing which is provided with two sight holes in diametrically opposite positions, the sight holes being fitted with $\frac{1}{2}$-inch plate glass to allow for observing both the condition of the bearing and the temperature of the oil by means of a thermometer hanging in the oil, illumination being secured by suspending an incandescent lamp within the casing. The stationary disk is fastened by steel dowel pins to a third disk which has a spherical seat scraped to fit a support which in turn is bolted to the thrust girder. This spherical disk makes it possible to take up slight deviations from the vertical, where a more rigid construction with the thrust-disk bearing might cause trouble. The radius of the spherical disk and ball seat is 3 feet 4 inches. The bottom of this is grooved at six points to allow the oil to pass from the outer or discharge chamber to the space where it can reach the bearing surfaces at the inner ring. As a result of the trials, all units were equipped with the combination bearing.

Brake. The brake on the shafts of the units in Power House No. 2 is hung from the thrust-bearing girder, and clasps a flanged collar, or brake wheel, bolted to the end of the adjacent section of the tubular shaft. It is operated either by compressed air at 100-pound pressure, or by hand. Its office, in addition to providing a means of stopping the machines quickly in case of accident, is to bring the rotating members to a stop on an ordinary shutdown, in a reasonable time. For example, without the brake, the machines would rotate for 30 to 45 minutes after the closing of the turbine gates; while, with the brake, full stop is obtained in less than a minute.

The brake consists of two levers on opposite sides of the brake

wheel, and a mechanism for drawing the levers toward each other to bring two oppositely located brake shoes to a bearing on the brake wheel. Each lever has the fulcrum at one end, the point of power application at the other, and the shoe at the mid-point. The shoes are lined with maple, and together have a bearing surface of

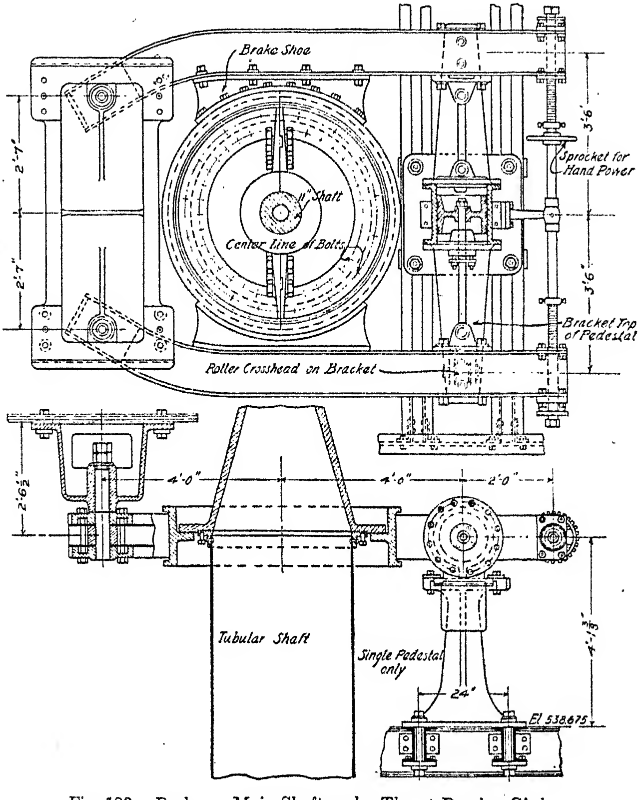


Fig. 193. Brake on Main Shaft under Thrust-Bearing Girders

180 degrees of the circumference of the wheel, or 7.2 square feet. The air piston has a diameter of 13 inches and a total stroke of 6 inches, and the force exerted to draw the levers together is multiplied by 2 at the shoes. The brake wheel is 5 feet in diameter, and 13 inches wide. The compressed air is controlled from the generator floor, as is also the hand operation of the brake. For the latter a sprocket-and-chain transmission turns a screw shaft to bring the levers together. Coil springs on the shaft keep the brake shoes normally free from the brake wheel, Fig. 193.

Balancing Piston. A special piston is here provided, 4 feet 6 inches in diameter, the water pressure on which, due to the head, partially balances the weight. The excess or unbalanced load in this case is taken by the oil bearing shown, oil being forced between the bearing surfaces.

Speed Regulation. In Power House No. 1, as originally installed, this was accomplished by means of a sensitive governor acting on a

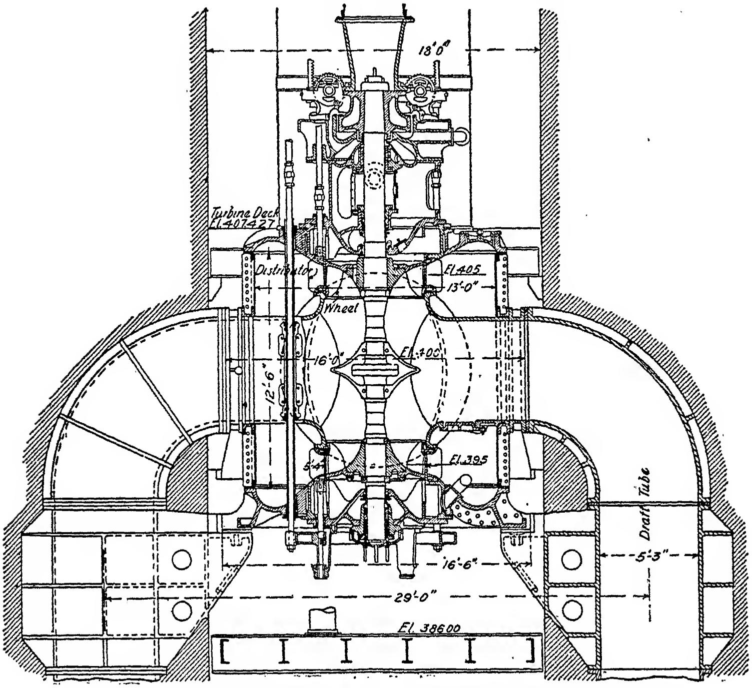


Fig. 194. Detail of 10,000-H.P. Turbine, Canadian Niagara Power Company

ratchet wheel connected with the sluices. In the later installations the relay is of the hydraulic type. In this case the sensitive governor in action opens a valve, thus actuating a ram driven by oil from an oil reservoir, at a pressure of 1200 pounds per square inch.

CANADIAN NIAGARA POWER COMPANY

159. Comparative Improvement. The installation of this company was designed to be worked in combination with that of the Niagara Falls Power Company, so that when necessary, it might assist the works on the other side of the river. The works were not

started on the Canadian side until it was thought that all the problems involved had been satisfactorily solved by experience on the American

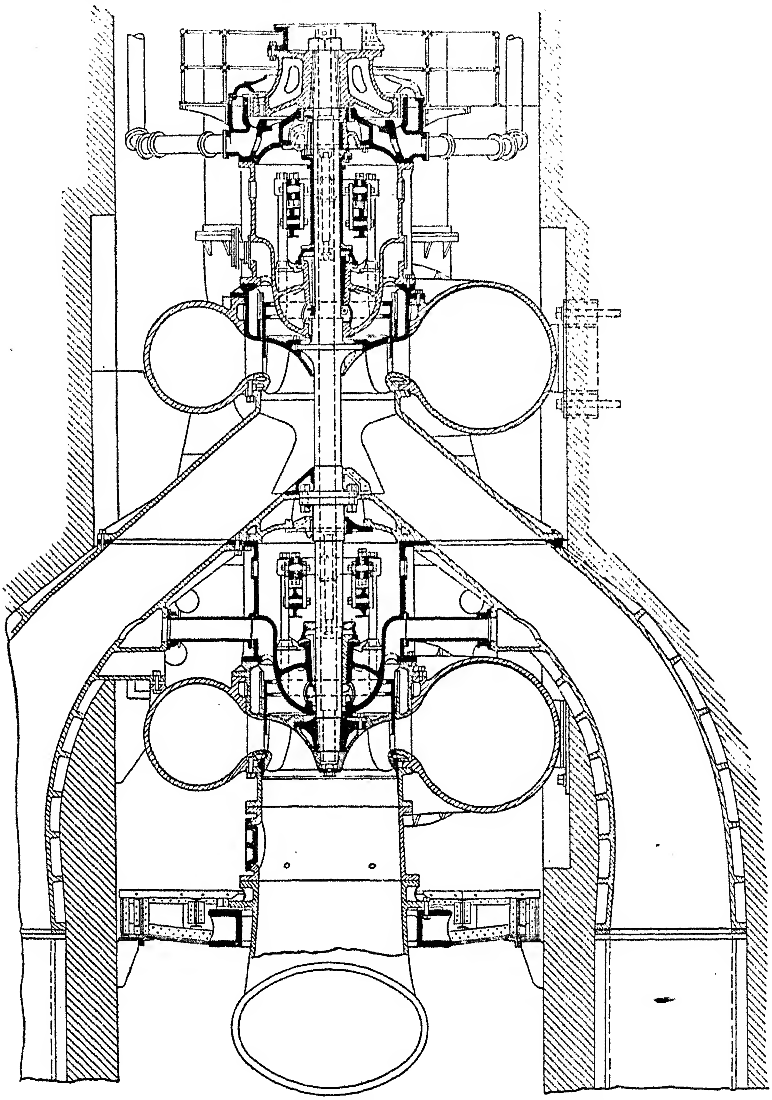


Fig. 195. Cross-Section of 12,500-H.P. Turbine for Canadian Niagara Power Company
Courtesy of "Electrical World"

side. When complete, the power house will contain eleven units, five of 10,000 horsepower and six of 12,500 horsepower, a total ultimate

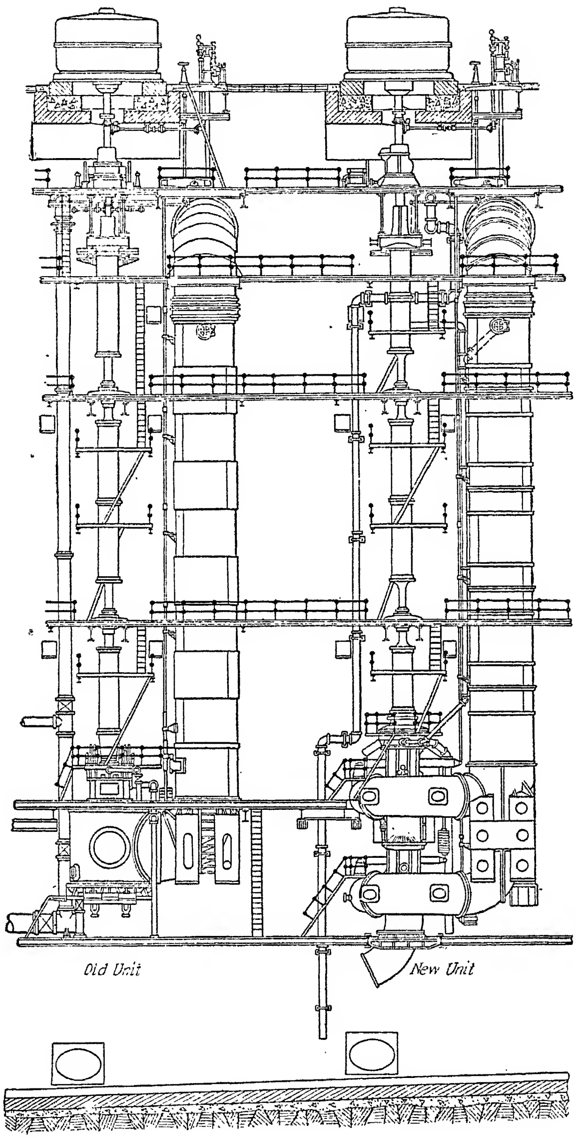


Fig. 196. Elevation of Old and New Units in Station of Canadian Niagara Power Company
Courtesy of "Electrical World"

capacity of 125,000 horsepower. The adoption of these large units resulted in a material reduction in the size of the wheel pit, canal,

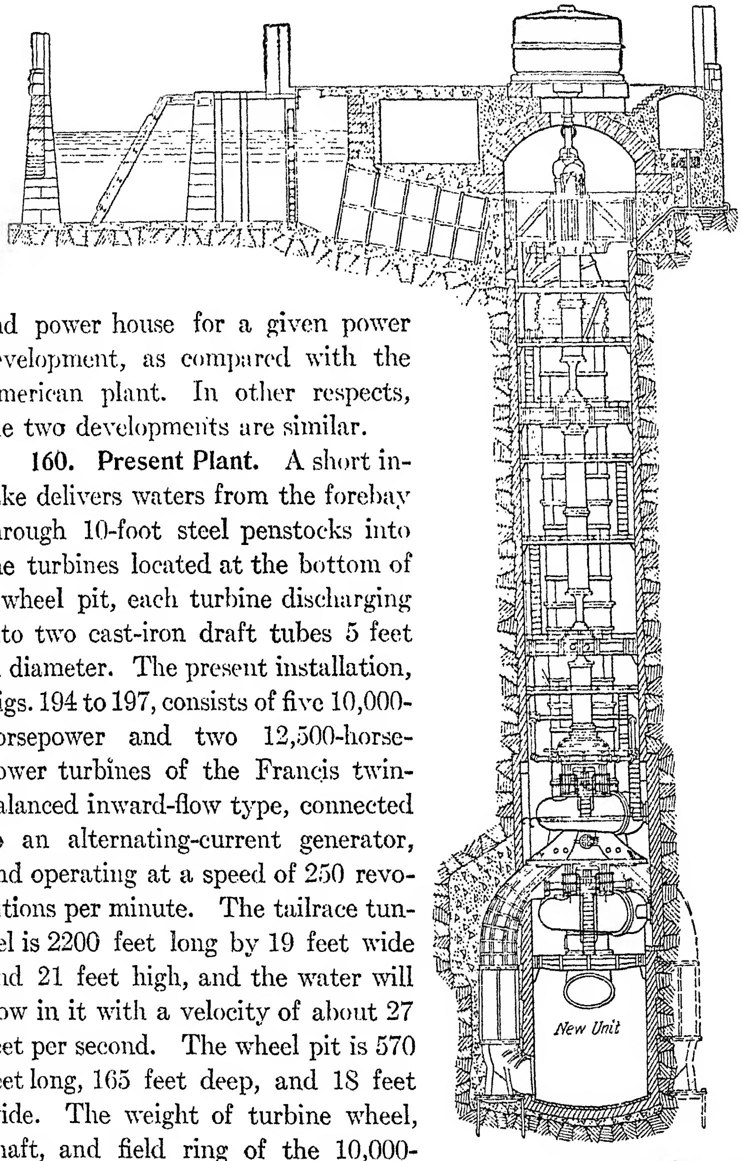


Fig. 197. Cross-Section of New Turbine Pit for Canadian Niagara Power Company

Courtesy of "Electrical World"

and power house for a given power development, as compared with the American plant. In other respects, the two developments are similar.

160. Present Plant. A short intake delivers waters from the forebay through 10-foot steel penstocks into the turbines located at the bottom of a wheel pit, each turbine discharging into two cast-iron draft tubes 5 feet in diameter. The present installation, Figs. 194 to 197, consists of five 10,000-horsepower and two 12,500-horsepower turbines of the Francis twin-balanced inward-flow type, connected to an alternating-current generator, and operating at a speed of 250 revolutions per minute. The tailrace tunnel is 2200 feet long by 19 feet wide and 21 feet high, and the water will flow in it with a velocity of about 27 feet per second. The wheel pit is 570 feet long, 165 feet deep, and 18 feet wide. The weight of turbine wheel, shaft, and field ring of the 10,000-horsepower units is 120 tons, and this weight is carried by a balancing piston

on which the water acts at the pressure due to the fall. Part of the current generated is transmitted to a transformer station on the bluff above the power house, and part is sent to the power station of the American Niagara Falls Power Company.

Turbine Efficiency. One of the new turbines, Fig. 195, installed in 1910, has shown a most gratifying performance. When operating under a head of 133 feet, it delivers 12,800 horsepower at the switchboard, and under normal running conditions the head and consequently the output of the turbine will be somewhat greater. The head was purposely reduced to 133 feet by means of the tailwater regulating gate at the lower end of the wheel pit so that the head during the test would approximate that which will exist when the ultimate development is reached. The indications are that the efficiency of the unit is equal to or better than that of the turbines of the older type, Fig. 194, while their cost is considerably less.

TORONTO POWER COMPANY, LTD.

(Formerly Electrical Development Company of Ontario)

161. Features of Construction. The success of the Niagara Falls Power Company stimulated other enterprises of similar magnitude. The Electrical Development Company of Ontario obtained water-power rights to erect an installation of 125,000 horsepower on the Canadian side, general plan, Fig. 176. In order to construct a masonry intake dam in the river, a cribwork cofferdam 600 feet long was built in some of the worst of the upper rapids, laying bare 11 acres of the river bed. This temporary dam was, at the worst part, in water 24 feet deep and flowing at probably 30 miles an hour. Its construction was an engineering feat of the greatest boldness. Within it was constructed a concrete gathering dam with granite coping, to direct water into the intake, while floating ice will pass over the dam and back into the river. The water, before entering the power house, must pass through submerged arches and screens. The tailrace tunnel is 26 feet high, $23\frac{1}{2}$ feet wide, and 1900 feet long. A driftway to the mouth of the tunnel was first driven, and then the tunnel excavated back from the mouth. The excavated material was thrown down from the mouth of the tunnel into the lower river, where it has disappeared. The wheel pit is 416 feet long, 27 feet wide, and 150 feet deep. A feature of this

installation is the two-branch tailrace tunnel, one branch on each side of the wheel pit, five turbines discharging into the one, and six into the other. These two branches unite into one large tunnel which is carried under the upper rapids of Niagara River to about the center of Horseshoe Falls, where it discharges into the lower river.

162. Power Equipment. There have already been installed four turbines of 12,000-horsepower capacity and three of 15,000 horsepower, or a total of 93,000 horsepower up to 1914; 4 additional units are contemplated. With full installation this station will have a capacity of 125,000 horsepower when using all the water granted by the treaty. The turbines are connected to alternating-current generators by hollow vertical shafts, 115 feet long, supported by 3 intermediate step bearings. The transformer and distributing station is situated on the bluff above the power house. The hydraulic units are of the vertical shaft double-runner central-discharge type, the four units being provided with cylinder gates and the three with movable guide vanes.

ONTARIO POWER COMPANY

163. Power Facilities and Equipment. The plans of this company differ essentially from the others, and are no doubt partly conditioned by the fact that the ground nearer the falls is already occupied. The power house is situated on the level of the lower river on the Canadian side, just below Horseshoe Falls. It is 76 feet wide and 1000 feet in completed length, the narrowness being due to the physical conditions of the location. The intake located at the top of the rapids on the Canadian side, near the Dufferin Islands, conserves the head of the upper rapids, and is especially designed with reference to ice difficulties.

The openings in the intake dam have a curtain dipping 9 feet into the water, below which the flow to the turbines takes place, the floating ice being carried past. A second curtain on the same principle is constructed between the forebay and inner basin, and the ice in the outer basin is carried forward over the lower part of the outer dam. The ice in winter is a serious difficulty at Niagara. Cake ice floats down from the upper lakes, and "mush" ice is formed in the turbulent rapids, primarily by the freezing of spray and foam. For ice in this latter form, there are screen frames.

From the intake, three great steel conduits, 18 feet and 20 feet in diameter, will convey the water round the other power houses to the top of the bluff below the falls. These conduits, of which two had been constructed by 1914, are of reinforced concrete, and of $\frac{1}{2}$-inch steel plates, stiffened with bulb irons and encased in concrete. The velocity in the conduits will be 15 feet per second. There is a spillway at the end, formed by a weir, to prevent water hammer in the pipes. The flow over the weir passes down through a helical culvert or spillway in the rock to the lower river. From the conduits the water will be taken down to the turbines through twenty-two steel pipes, 9 feet in diameter, passing down the face of the bluff.

The power house is on a platform at the foot of the bluff, and just above the level of the lower river. Each turbine unit consists of 2 Francis mixed-flow wheels discharging from the inside at the center so as to balance the horizontal thrust. They operate at a speed of 187.5 revolutions per minute under an effective head of 175 feet, including a 20-foot draft-tube head. The turbines were placed thus high above the river to provide for excessive variations of level at this site. Each pair of turbines is coupled by means of a horizontal shaft to an alternating-current generator.

SUMMARY OF ELECTRICAL DEVELOPMENTS

Capacities of Large Power Plants at Niagara Falls

Plant	Capacity	
	Present Installation (hp.)	Ultimate Contemplated (hp.)
AMERICAN SIDE		
Niagara Falls Power Co.		
Power House No. 1	55,000	55,000
Power House No. 2	60,500	60,500
Hydraulic Power Co.		
Power House No. 1 (abandoned)		
Power House No. 2	35,000	35,000
Power House No. 3	120,000	130,000
CANADIAN SIDE		
Canadian Niagara Power Co.	75,000	125,000
Ontario Power Co.	140,000	166,000
Toronto Power Co., Ltd.	93,000	125,000
Total	578,500	696,500

The first three units have a capacity of 10,000 horsepower each; the next seven units a capacity of 12,000 horsepower each; the last two, 13,000 horsepower each. The present installation of twelve units has an aggregate capacity of 140,000 horsepower. Two additional units of 13,000 horsepower each are being added, which will raise the total capacity of this station to 166,000 horsepower. The transformer and distributing station are on the bluff above the power house.

164. Present and Prospective Usage. The amount of water thus diverted from the falls amounts at the present time to about 17 per cent of the total quantity passing over the falls.

Several plans have recently been proposed for utilizing the head now lost in Lower Rapids, between the base of the falls and Lewiston, without diverting any additional water from the falls. One of these plans contemplated the excavation of a surface canal between a site near the Grand Trunk Railway bridge at the head of Whirlpool Rapids and Devil's Hole on the American side, about 10,000 feet downstream, thus converting into power an available head of about 73 feet, which is now wasted.

PACIFIC SLOPE DEVELOPMENT

SNOQUALMIE FALLS POWER COMPANY*

165. General Characteristics. Within the past few years a number of plants have been established on the Pacific slope, to utilize natural water powers for generating electricity to be transmitted to distant points, and there used for lighting and power purposes. Among the most interesting and important of these plants is that at the Snoqualmie Falls, in Washington. For this plant, no long flume or pipe line is required to develop the necessary head of water, as the Snoqualmie River has at the falls a vertical drop of 270 feet, giving an available energy of 30,000 to 100,000 horsepower. In this respect the plant resembles those at Niagara Falls. In the placing of the electrical machinery, however, there is an essential difference; for, while the Niagara Falls plants have this placed in a building above ground, the Snoqualmie Falls plant has the water wheels and electrical machinery all installed together

* *Engineering News*, December, 1900.

in a large underground chamber whose floor is directly above the tailrace tunnel, which extends to the river below the falls. The force of the water is used to drive impulse wheels on horizontal shafts, instead of turbines on vertical shafts, as at Niagara Falls. Another notable feature of the plant is the use of aluminum wire for the long-distance transmission lines. The entire plant represents an investment of about $1,000,000.

The great fall of the Snoqualmie River is about 34.5 miles northeast from Tacoma in a straight line, the same distance southeast from Everett, and 25 miles west from Seattle, being situated in the foothills of the Cascade Range. The river proper commences about 3 miles above the fall, at the junction of three forks which flow westward down the slopes of the range. Below the fall the river runs almost due north, and finally flows into Puget Sound near the city of Everett. The flow of the river is about 1000 cubic feet per second at its lowest stage, increasing to over 10,000 cubic feet per second at its flood periods. The river does not freeze during the winter, and there is neither floating ice nor anchor ice to be dealt with.

An investigation showed that by the construction of dams or dikes, some of the large lakes on the watershed could be utilized as impounding reservoirs, so as to insure a uniform flow sufficient to develop nearly 100,000 horsepower throughout the year, should a demand for so much power eventually be found. It was also determined that by the erection of a 50-foot dam above the headworks, a reservoir could be formed, having an area of 15 square miles and an average depth of 25 feet. This would almost double the power, should the growth of the industries served make this desirable in the future.

The rock at the falls is basaltic, with no regular cleavage. It is hard and nonabsorbent, and is apparantly divided by seams into great ledges. These conditions led to the adoption of the plan of placing the machinery in an underground chamber, as already noted. It was at one time proposed to build a power house near the base of the fall; but this would have been at a disadvantage, on account of the clouds of spray, which keep everything damp, and which coat all of the surroundings with ice in cold weather.

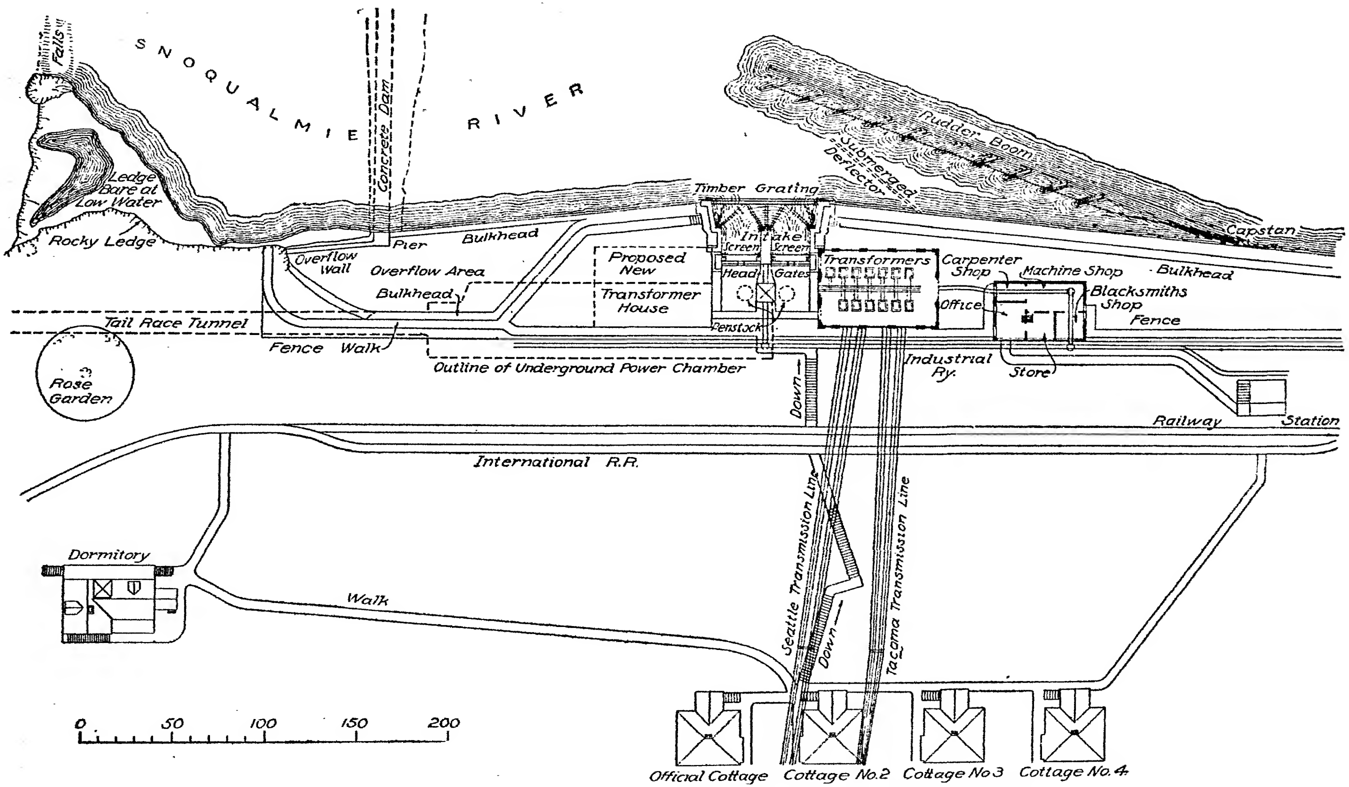


Fig. 198. Plan of Headworks of the Snoqualmie Falls Power Company, Washington

166. Features of Plant. *Headworks.* A plan of the headworks above the falls is shown in Fig. 198, which also shows the position of the underground power chamber and the tail-race tunnel. The intake bay is a rectangular chamber, about 60 feet long parallel with the river, and 20 feet wide. It has its walls, and a center pier of concrete masonry 6 feet thick and 25 feet high, built upon solid rock formation, its floor being on a submerged reef about 5 feet above the river bed. This bay is protected from the river by a timber grating across the opening, supported by a steel-girder construction bearing against the walls and pier. The timbers are 12- by 12-inch, laid horizontally with 12-inch spaces between them, through which the water flows into the intake. This grating protects the works from floating trees and logs; while just inside the intake are inclined steel screens made of flat bars on edge, which serve to exclude the smaller débris. The intake thus has two headbays, separated by the pier.

A rudder boom 300 feet long is moored above the intake, and extends beyond it. By turning the capstan at the head of the boom, the rudders are thrown out, and cause the boom to swing out into mid-stream, so that it serves as a fender to deflect floating logs, etc., from the intake. The river is 150 feet wide from the headbay to the opposite shore, and about 15 feet deep at ordinary stages. The face of the intake was continued 400 feet upstream and 200 feet downstream, in the shape of heavy retaining walls built of sawed cedar timber, tarred. The space behind them is filled with excavated rock, and has a top dressing of soil for a lawn and shrubbery.

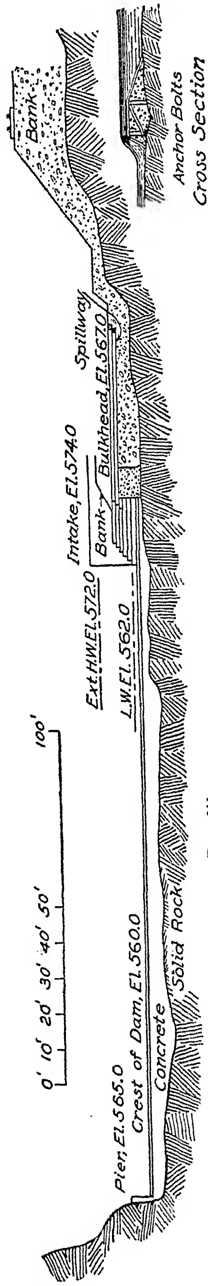


Fig. 199. Section Showing Submerged Dam of the Water-Power Plant of the Snoqualmie Falls Power Company, Washington

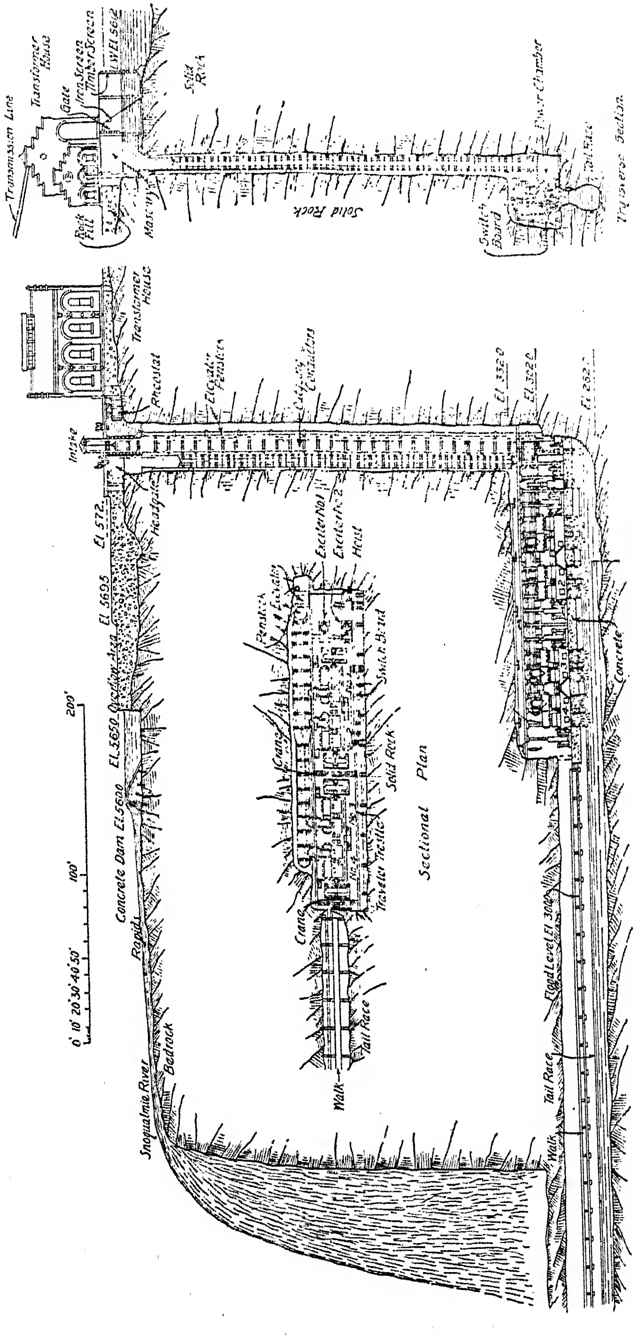


Fig. 200. Sectional Elevations of Plant of Snoqualmie Falls Power Company, Washington

At the end of the lower bulkhead, a submerged concrete dam is built across the river, resting on the rock bottom, and this raises the low-water elevation of the river 6 feet at the intake. This dam, whose location is shown in the plan, Fig. 198, is of the form shown by the elevation and section, Fig. 199. It was first framed of heavy timbers sheeted over with 6-inch planking, and then filled in solid with concrete. It was built about the time of low-water flow, portions of the river bed being laid bare by cofferdams. Preparatory to the construction of the dam, the river bed was thoroughly cleaned of loose rock, and was roughened by occasional blasts so as to afford a good footing. In addition to this, pieces of steel rail were driven in holes drilled 2 feet deep, the rails extending up into the concrete body of the dam. Old railway cables were also embedded in the concrete to perfect the bond. The dam has a batter of 2:1 upstream, and $\frac{1}{2}$:1 downstream, with a level crest 8 feet wide. At each end of the dam is an abutment pier of concrete 8 feet square, these being 210 feet apart. The dam was built on natural rock ledge, about 3 feet above the river bottom. It is always submerged from 2 to 10 feet, according to the stage of the river, and varies in height from 3 to 10 feet, and in width on bed rock from 16 to 35 feet, according to the conformation of the river bottom. The lower bulkhead is only 5 feet higher than the dam, so that flood waters have a very considerably increased sectional area of discharge, as shown by the plan. The capacity of this spillway is such as to insure the complete discharge of an extreme flood without the river backing up to an unusual elevation. The top of the upper bulkhead is above flood level.

Underground Power Chamber. The general arrangement of the plant, with its underground power chamber, is shown in Fig. 200. About 300 feet above the falls, a shaft 10 feet by 27 feet was sunk in the bed of the river on the south side, descending 270 feet to the level of the river below the falls. While this shaft was being excavated, a tunnel 12 feet wide and 24 feet high, with a fall of 2 feet in its entire length, was drifted in from the face of the ledge below the falls, to an intersection with the bottom of the shaft, a distance of 650 feet. Beginning at the foot of the shaft, and extending over and along the tunnel, a chamber 200 feet in length, 40 feet wide, and 30 feet high, with the floor at the elevation of

high water below the falls, was excavated out of the solid rock, Fig. 201. This chamber forms the power house, or machinery room, in which the water wheels and electric generators have been installed, Fig. 202. At average stages of the river, the water is about 12 feet deep in the tunnel; while during flood seasons the tunnel is nearly filled. The tunnel extends under the floor of the chamber, forming a tailrace with concrete roof 5 feet thick. The walls of the chamber have been left rough and whitewashed, while the floor is covered with concrete.

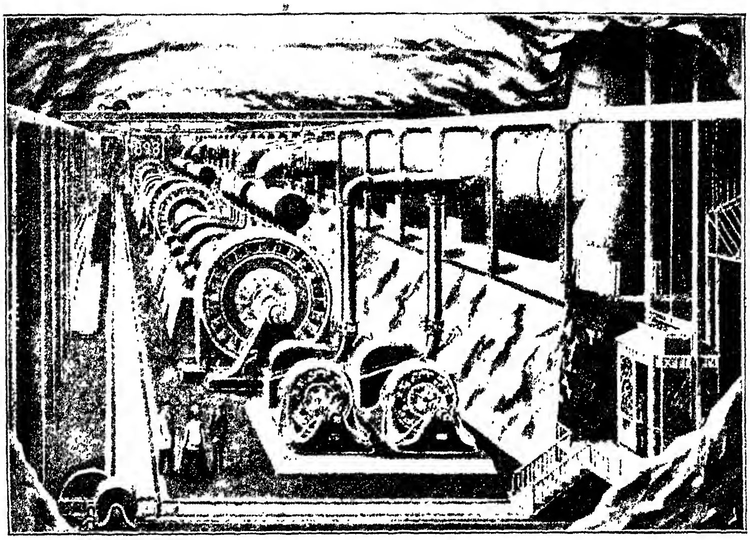


Fig. 201. Interior of Power Chamber at Snoqualmie Falls

The chamber is ventilated by natural draft through the tailrace and up the shaft, the draft being so strong that it was necessary to curb it. The chamber is said to be cool and perfectly dry, the temperature remaining the same (about 55° Fahrenheit) throughout the year. This low and uniform temperature contributes to the high efficiency of the generators.

Water Supply. Each intake bay contains a massive head gate, moving vertically which controls the flow of water through an opening 8 by 12 feet through the shore wall into the penstock. The gate is raised and lowered by mechanism connected with the piston rod of a hydraulic cylinder. The shaft is 10 by 27 feet, and at the

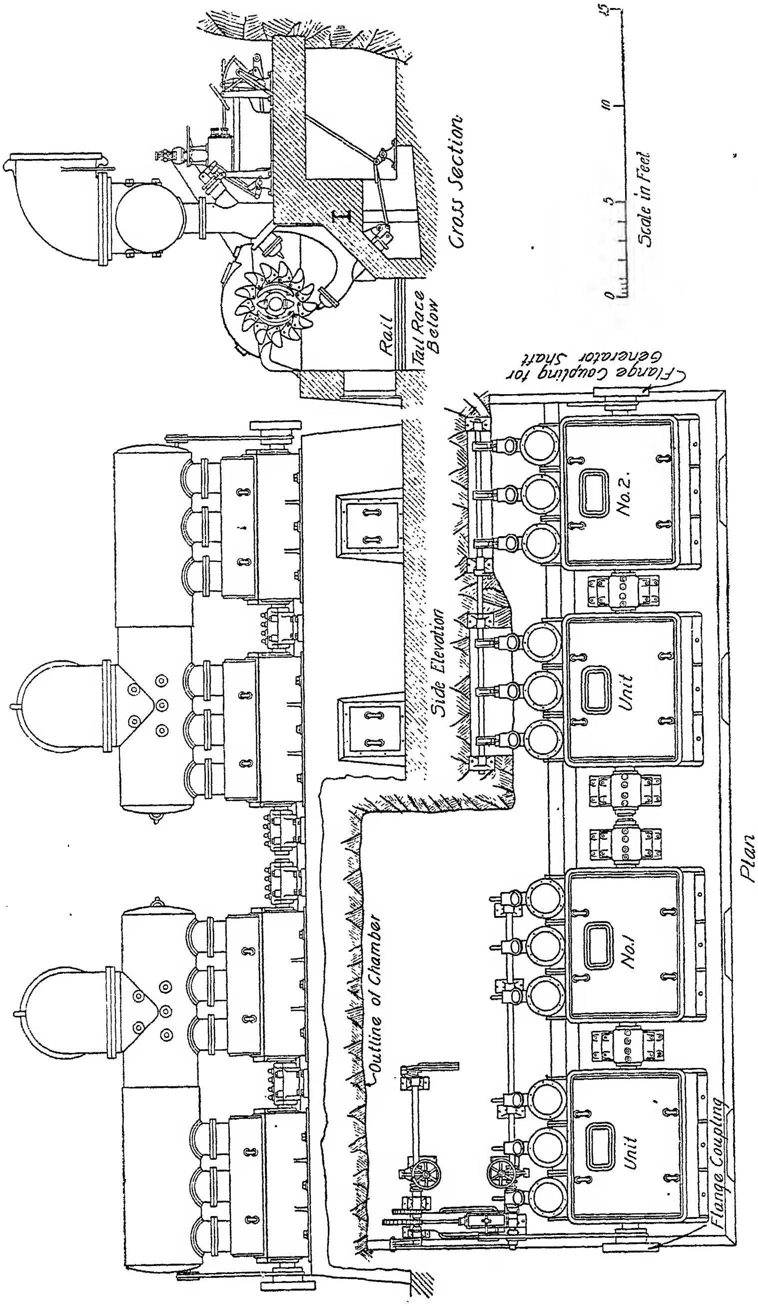


Fig. 202. Elevation, Plan, and Section of Water Motors in Plant of the Snoqualmie Falls Power Company, Washington

top has three compartments; the two end compartments are for the penstocks; while the center one, enclosed at the top by a steel bulkhead, forms a shaft 8 by 10 feet for the hydraulic elevator and the main cables forming the outgoing conductors, and also for raising and lowering machinery, etc. The steel bulkhead which encloses this center shaft extends from the bottom of the intake bay to the surface of the ground. It is built up of steel plates, and is stiffened by horizontal frames of I-beams on the outside, riveted to the plates and to each other at the ends. Below it, the elevator shaft is timbered and sheathed with plank. At the surface, this shaft is surmounted by a small building. The penstock first built is of steel pipe $7\frac{1}{2}$ feet in diameter, passing through a concrete roof which keeps the shaft water-tight. The plates are in 8-foot courses, and are 1 inch thick for the lower half of the pipes; in the upper half, the thickness decreases from $\frac{7}{8}$ to $\frac{1}{2}$ inch at the top. The joints are heavily riveted, and calked water-tight. At a depth of 250 feet, the penstock reaches the chamber, and connects with a horizontal cylindrical receiver which rests on a rock bench in the north side of the chamber, 12 feet above the floor. This receiver extends almost the full length of the chamber. Its diameter is 10 feet for half its length, and then reduces to 8 feet. It is built up of 1-inch plates 8 feet wide. The penstock and receiver weigh 225 tons, and the weight of the water column in the penstock is 340 tons. A small independent penstock supplies water to the elevator machinery, as shown in Fig. 200.

At four points in the length of the receiver are 4-foot branches extending from the side, each branch being fitted with a gate valve. These valves weigh 23,000 pounds each, and are said to be the largest valves in the world operated under such high pressure. Each branch has a cast-iron elbow turning downward and opening into the horizontal cylindrical receiver of a water motor. These elbows have an inside diameter of 4 feet, and the metal is 2 inches thick, each casting weighing 8000 pounds. Owing to the size and form, special care had to be taken to avoid internal strains in the castings which might lead to rupture, especially in view of the high pressure which they have to withstand.

Generating Units. The main generating plant consists of four electric generators, each driven by a 45-inch Doble water motor of

2500 horsepower, coupled directly to it. Each motor consists of a shaft carrying 6 tangential-jet wheels, with two nozzles to each wheel. Each of the four elbows above referred to is bolted to a flanged ring on a horizontal cylindrical receiver, 48 inches in diameter and 20 feet 8 inches long. This is made of two $\frac{1}{2}$-inch steel plates 10 feet wide, and of sufficient length to make the shell with only one longitudinal seam, which is double-riveted. The heads are of dished steel plates. The receiver is supported by six pipes, each of which carries two nozzles delivering jets at right angles to each other, the nozzles entering the side and bottom of the wheel casing. The use of the receiver effects an even distribution of the flow from the elbow to the several nozzle-pipes, and also a steady and uniform rate of flow. From the buckets of the motors, the water falls through draft openings in the floor directly into the tailrace channel.

To handle the volume of water necessary to develop the power in each unit, requires 12 jets $3\frac{7}{8}$ inches in diameter, discharging against six wheels. For convenience of bearing and shaft design, these wheels are divided into two groups of 3 wheels each, each group being in a separate housing, with a bearing between. This arrangement makes two groups of 3 vertical nozzle pieces each.

One of the special features of this plant is the needle-regulating tips used on the nozzles. They not only throw a perfect and unbroken stream, but give absolute control over the quantity of water applied to the wheels, and therefore over the power output of the unit. As these tips are controlled by the governor, the arrangement gives an excellent degree of speed regulation with variable load, at high efficiency. The full size of the jet is 3 inches; it is a solid, smooth stream, delivered with a head of 253 feet, entirely free from swirling or other disturbance. This form of nozzle maintains the same condition from full-jet size to $\frac{1}{10}$ of the jet area. The regulating nozzles are operated from two long rocker shafts, one controlling the upper, and the other controlling the lower nozzle tips. Both rocker shafts are operated by a Lombard governor, which is connected to the rocker shaft by cranks and connections so arranged with clutches that either or both rocker shafts can be disconnected from the governor and operated or regulated by the hand wheel on the pedestal stand. By this governor arrangement, with the regulating nozzle tips, the wheels use water in proportion to the power

developed, so that they operate with high efficiency at part as well as at full load.

The wheels are encased in sheet-steel housings with cast-iron fronts, 3 wheels in each, so that there are 2 housings to each unit. The housings are made with the upper half removable, to provide access to the wheels when desired. The cast-iron front of each housing is made of such form as to provide a deflector guard, which takes care of the water thrown from the wheels by the centrifugal action, and directs this water into the tailrace, and thus prevents its being driven around the housing by the air currents created by the rapidly revolving wheels. In the top housing is a guarded opening to permit the indraft of air to replace that driven out of the housing and down the tailrace by the rush of the water and the action of the wheels as centrifugal blowers. To prevent water from splashing out where the shaft passes through the side of the housing, the opening is protected with patented centrifugal disks and guard-frames. Although this arrangement accomplishes its object in preventing the outflow of water, it also permits a large and free indraft of air at this point to replace that driven out by the action of the wheels and the water.

Each wheel unit weighs about 100,000 pounds, in addition to the weight of water in the distributing receiver and the nozzles; and in view of the high speed of the parts, and the power developed, careful design and construction were required for the foundations, which are of concrete, built solidly into the floor and one side wall of the chamber; and the lower part of the steel wheel housing is firmly built into the concrete walls. The waste water drops from the wheels directly into the tailrace.

A tunnel is provided under the governor platform for the lower rocker shaft and connections that operate the adjustable tips of the lower nozzles, and thus makes this operating gear accessible. The foundation for each unit is divided into two compartments corresponding to the two wheel housings, and a 2- by 3-foot doorway is formed in the front wall of each. Four steel rails are built into the concrete across the opening into the tailrace, to support a temporary floor, when it is desired to enter the foundation for the purpose of inspecting the wheels or nozzle tips without removing the top wheel housing.

167. Additional Construction. *In 1905 a second penstock, similar to the earlier one, was installed, connected directly at the bottom with a 10,000-horsepower radial-inflow Francis turbine. This unit has a horizontal-shaft spiral-case single 66-inch runner operating at 300 revolutions per minute, discharging axially into a single draft tube. The tangential units are governed by Lombard type "L" governors acting on needle nozzles, and the turbine by Lombard type "N" governor acting on the turbine gates. The exciter equipment consists of two 125-horsepower 300-revolution-per-minute Pelton impulse wheels. The tail race extends the entire length of the cavity under the machinery floor and empties into the tailrace tunnel.

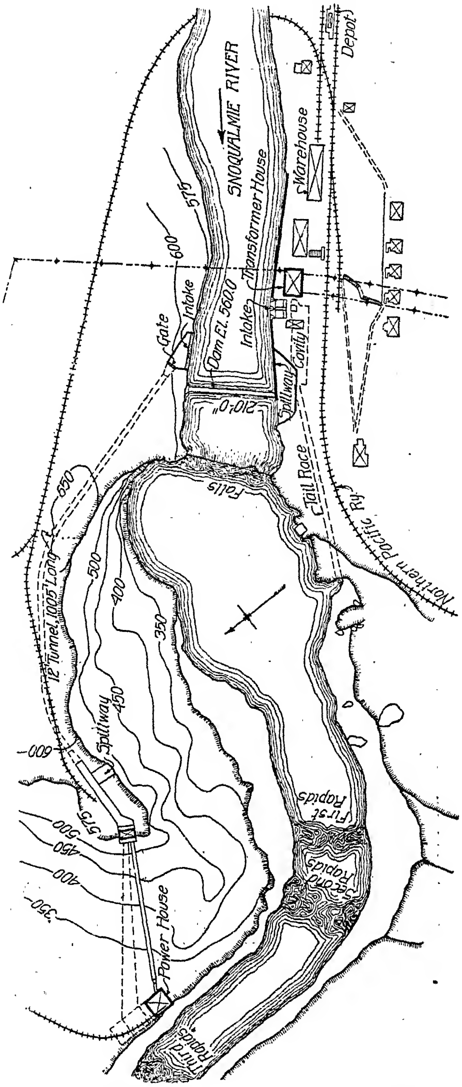


Fig. 203. Map of Snoqualmie River District Showing Location of Old and New Plant
Courtesy of "Engineering Record"

The above described installation at Power House No. 1 filled the cavity, and when the need for more capacity became imperative it was found to be inadvisable to attempt to enlarge the old plant by further rock excavation, as a large amount of blasting would have been necessary

* Magnus T. Crawford, electrical engineer, *Engineering Record*, January 13, 1912.

in close proximity to the machinery. While the underground plant had the advantage of absolute protection from weather

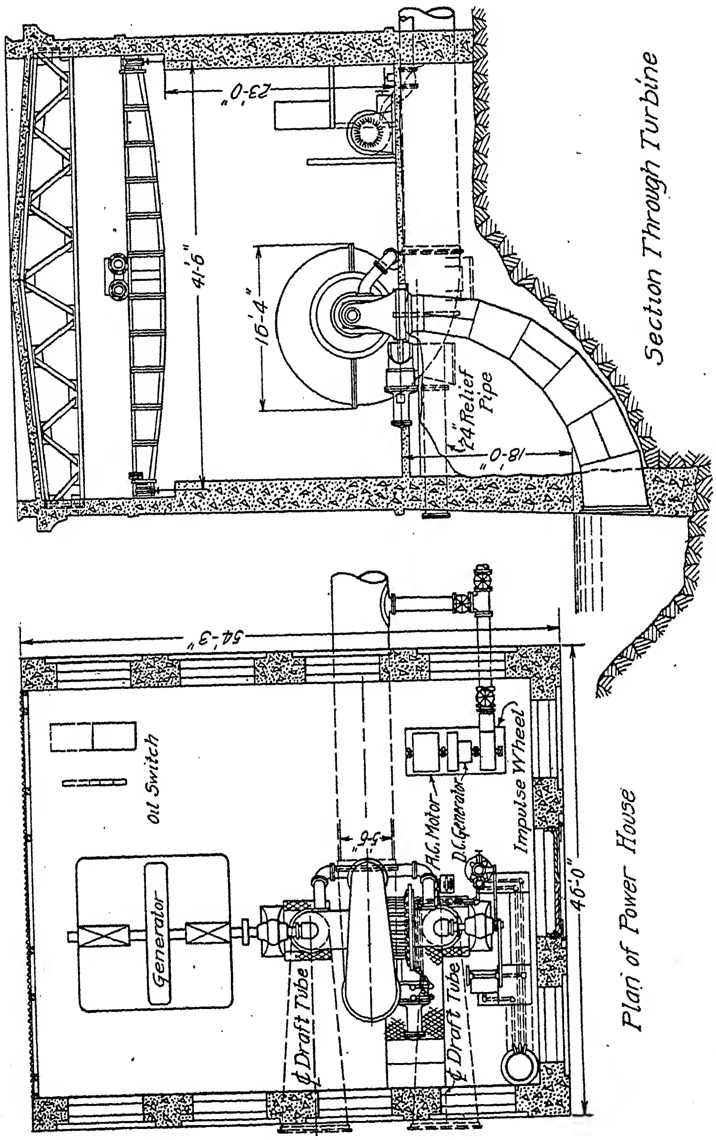


Fig. 204. Plan and Section of Equipment in New Power House of Snoqualmie Power Company, Washington
Courtesy of "Engineering Record"

conditions, it practically barred extensions and made alteration or repair work diffcult. An appreciable amount of head was

lost in the long tailrace and rapids just below the falls. In view of these and other considerations, the installation of an entirely separate station below the falls and around the bend of the river was recommended. This station, which is shown in the map, Fig. 203, and in the plant sections, Fig. 204, is arranged for the future addition of two 10,000-horsepower turbines.

Headworks of Power House No. 2. At a point on the west bank directly across the river from the intake to the old power house the new intake has been built with a width of 85 feet between two concrete bulkheads, with a concrete pier 5 feet wide in the center.

Leaves and sticks are kept out by screens made of $\frac{1}{4}$ by 4-inch rack bars spaced 1 inch on centers. The tunnel leads off from this intake at an angle of about 30 degrees with two gates 10 feet wide at the mouth of the tunnel. Rudder booms made of 12- by 12-inch timbers, 150 feet in length, are floated in the stream in front of the intake and fend off large logs at high water. Concrete walls 2 feet thick form the sides of the intake between the gates and the bulkheads.

From the intake a circular tunnel of 12 feet net diameter has been excavated through the rock a distance of 1035 feet, connecting with an open rectangular forebay channel 220 feet long and 20 feet wide leading to the gate house at the head of the penstocks. With 1340 second-feet of water at a velocity of 11.85 feet per second, the loss of head in the tunnel is 5 feet. A spillway 30 feet wide has been excavated from the forebay to the edge of the cliff discharging over the side below the falls.

The intake, tunnel, forebay, and gate house are constructed for an installation of three 10,000-horsepower units at Power House No. 2, which, with the present installation of 20,000 horsepower at Power House No. 1, will make a total of 50,000 horsepower in the development planned at present.

The gate house is a small concrete building containing the electrically operated headgates. These are controlled from a special panel in the power house, which indicates the position of the headgate at all times.

The penstock is built up of riveted steel plate, with an inside diameter of 7 feet, resting on concrete anchorages on the bed rock. The steel plate varies in thickness from $\frac{3}{8}$ inch at the top to $\frac{7}{8}$ inch

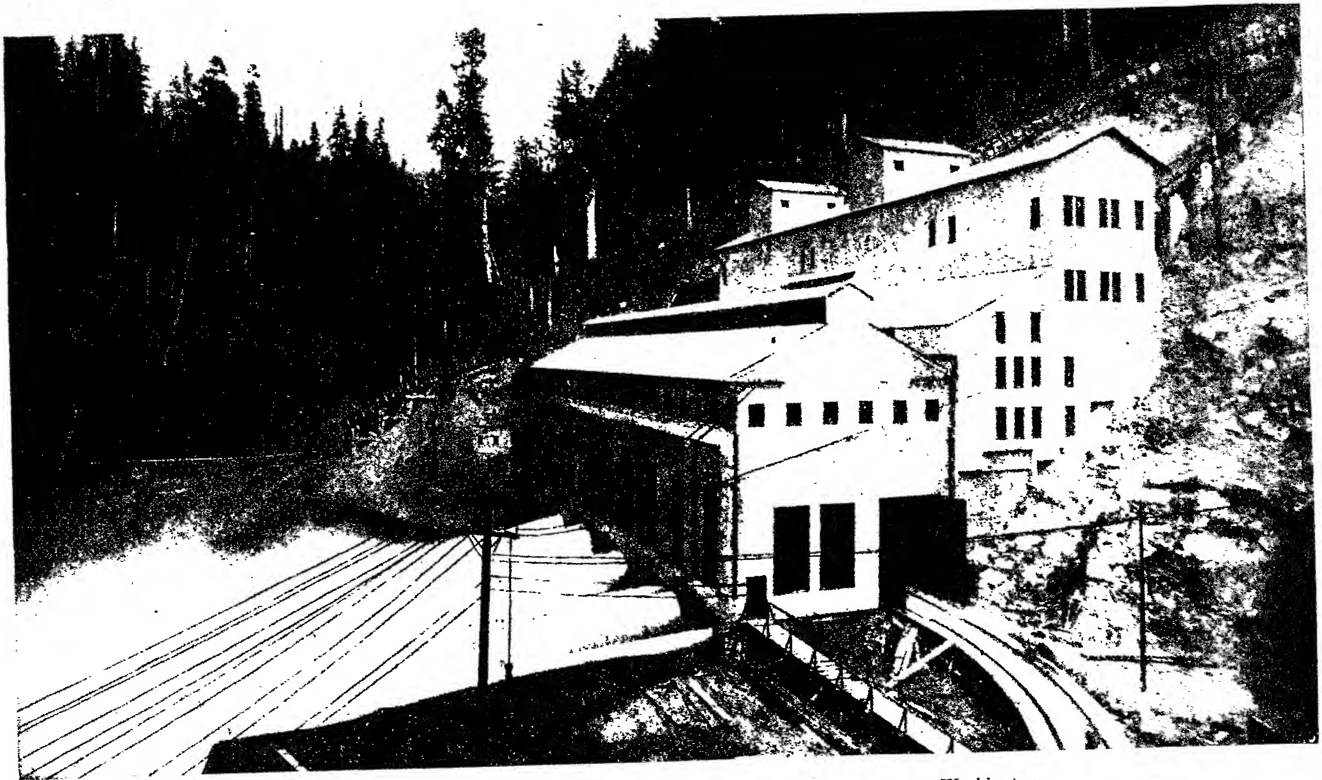


Fig. 205. Power Station of the Puget Sound Power Company at Electron, Washington

Located 48 miles from Seattle, 32 miles from Tacoma, on the Puyallup River, which originates in the glaciers of Mount Rainier (Tacoma). From the reservoir up the mountain side, fed by a flume 10 miles long, water under a head of 865 feet is delivered to the Pelton water wheels connected to the generators, through riveted-steel penstocks (one of which is seen at the right). Power is transmitted to Tacoma, Seattle, and other points, for street-car, lighting, and other purposes.

at the receiver. The penstock has a total length of 466 feet in the vertical fall of 255 feet, making two slight bends.

The power house is a reinforced-concrete building of standard design.

Machines and Equipment. The water wheel has a guaranteed efficiency of 60 per cent at 3000 horsepower, 72 per cent at 5000 horsepower, and 82 per cent at 8000 to 10,000 horsepower. Under test it developed 12,500 horsepower at 361 revolutions per minute, with 109.5 pounds on the pressure gage and full-gate opening. The wheel is of the Francis type, radial inflow with double axial discharge and horizontal shaft, with two bearings. The double axial discharge has been found to be very satisfactory, eliminating the troubles from thrust bearings. The inflow of water is controlled by swivel gates which are also guide vanes, directing the water against the runner blades. These swivel gates are connected to a common ring, which is actuated directly by the governor. At light loads these gates are nearly closed, the water spouting through a narrow slit between the guide vanes and striking the vanes in the runner, so that the turbine is virtually an impulse wheel under these conditions. At ordinary loads the gates are open and the wheel runs as a reaction turbine.

The governor is of the oil-pressure double floating-lever type, and its motion is transmitted through a pilot valve and relay valve so as to admit oil under high pressure to the cylinder actuating the swivel-gate ring. A 24-inch relief-valve pipe was furnished but is equipped with a blank flange, as no trouble is experienced from water hammer in the short penstock line.

PUGET SOUND POWER COMPANY

168. General Facilities. This hydroelectric plant at Electron, Washington, Fig. 205, is located on the Puyallup River, 32 miles from Tacoma and 48 miles from Seattle. This river has its origin in the glaciers and snow peaks of Mt. Rainier, the highest mountain in the United States; consequently an unfailing source of water is assured from the melting snow and ice.

The water-power scheme consists of the diversion of the Puyallup River by a low dam 200 feet long and 5 feet high, through a masonry intake perpendicular to the dam, and the conduction of its flow

by means of a flume 10 miles long to a reservoir and settling basin located on a high plateau, and thence by steel pipes to the Pelton wheels, which operate under a head of 865 feet.

The flow of water into the flume is controlled by a radial gate at the junction of flume and intake. The flume is 8 feet by 8 feet in cross-section, carried on trestle work, and provided with sand boxes, automatic spillways and emergency gates. The steel pipes taper from 48 inches at the upper ends to 36 inches at the lower. The flume and reservoir are constructed with a view to the ultimate development of 60,000 horsepower.

169. Power Units. The equipment installed in 1908 consists of four direct-connected Pelton wheels, each driving a 3500-kilowatt General Electric generator at 225 revolutions per minute; and two Pelton wheels, each direct-connected to a 150-kilowatt exciter. Each wheel unit has an overload capacity of 7500 horsepower, making the present output of the station 30,000 horsepower. This power is transmitted to Tacoma and to Seattle, being used for the various industrial enterprises in that section, and particularly for operating the extensive system of suburban electric roads in the vicinity of Seattle.

The ultimate installation is to consist of 8 units, and the entire equipment is so arranged as to provide for complete pilot control of both water wheel and electric apparatus, which is accomplished from the switchboard at one end of the building. The complete equipment of 8 units will require a building considerably over 200 feet in length. It was found necessary to reduce the length of each unit to the minimum, and for this and other reasons the Pelton "double-overhung" construction was adopted. This construction consists of one water wheel overhanging each end of the shaft beyond the bearings, of which there are two for each unit, placed one on each side of the engine-type generator, which is in the center of the bed plate; it enables bearings to be more nearly equally loaded than if entire output were obtained from one wheel. Each of the two wheels on each unit is required to develop 3750 horsepower capacity when operating under an effective head of 865 feet, at 225 revolutions per minute. Each wheel is provided with a 24-inch single-disk bronze-mounted gate valve, with 5-inch by-pass arranged for operating normally by electric motors from the switchboard,

and also provided with gear and worm wheel for quick and slow motion by hand. There is one combination needle and deflecting nozzle, with ball-and-socket joint, for each wheel, the weight of the swinging portion being suitably counterbalanced, and the position of the needle controlled by a hand wheel.

Control. The power developed on each wheel is controlled in two different ways: (1) by the deflecting portion of the nozzle which is actuated by an automatic governor, thus limiting the quantity of water impinging on the wheel; and (2) by varying the flow of water through the nozzle by means of the needle device above mentioned. Sudden changes of load are taken care of by means of the governor and deflecting nozzle; and consequently there is no variation in velocity of water in the main pipes; hence, no danger of water ram. The adjustment of the needle to vary the flow through the nozzle is a comparatively slow operation and therefore cannot injure the pipe line.

PACIFIC COAST POWER COMPANY

170. General Conditions.* The White River development of the Pacific Coast Power Company is located in the Puget Sound district of the State of Washington, some 25 miles southeast of Seattle and 10 miles east of Tacoma. It has recently been placed in operation and serves a double purpose; the two older water-power developments supplying this region, one on the Puyallup River and the other at Snoqualmie Falls, have an aggregate capacity of 35,000-kilowatts, and not only has this capacity become outgrown, but also neither plant is so situated as to have the advantage of extensive storage. The White River development has now added outright 36,000 horsepower to the generating capacity of the system, and has provided moreover an enormous storage capacity.

The White River rises in glaciers on the north and east slopes of Mt. Rainier and drains about 400 square miles. The minimum recorded flow at the point of diversion is 420 cubic feet per second, with a low-water period of 2 months in the fall, previous to the rainy season.

The scheme of development permits enormous storage, warranting an ultimate capacity of 100,000 horsepower. The headworks, flume, storage basins, and canals have been built for this capacity,

* *Engineering News*, April 11, 1912.

but the initial turbine installation is of 36,000 horsepower, with room for another 36,000 without change.

The point of diversion is near the town of Bulkley, Fig. 206, which is some 22 miles from Tacoma and 7 miles southeast of Lake Tapps. There is a high plateau between the point of diversion and Lake Tapps, slightly shelving and extending westward with a series of comparatively small natural depressions which have been converted into storage basins and connected by flumes and canals.

Lake Tapps is 2 miles from the lower valley of the White River and 450 feet above it, with a ridge 600 feet high between the lake and valley. The natural level of the lake has been raised 35 feet by dams and embankments and the flooded area has been increased from 600 to 3000 acres with an available storage of 2,225,000,000 cubic feet. The outlet canal extends from the northwestern arm of Lake Tapps to the western ridge of the plateau. A tunnel carries the water under the highest part of the ridge, and pipe lines lead down to the power house at Dieringer.

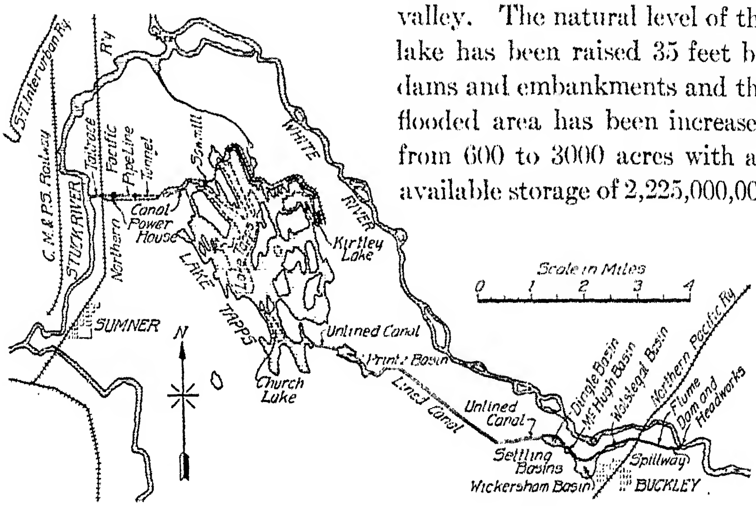


Fig. 206. Map Showing White River Power Development of Pacific Coast Power Company
Courtesy of "Engineering News"

171. Water Supply and Disposal. *Headworks.* The diversion dam is a special type for its function, to preserve the level of the river bottom and protect it against washing out at the point of diversion. At flood stages the river carries heavy drift so that the dam has been made low and broad with a surface designed to pass drift without injury. It is a rock-filled crib on concrete foundations. The entire length of 352 feet is a spillway capable of discharging

50,000 cubic feet per second—twice the quantity of the greatest recorded freshet. In times of low water, flashboards will raise the level 7 feet; these are easily removed and in case of sudden floods or heavy drift would be swept automatically without damage to the permanent structure. The concrete foundations rest on impervious hardpan. Concrete wing walls extend upstream on both sides of the dam, to prevent the stream from changing its course. The intake is of concrete with its opening perpendicular to the axis of the dam and parallel with the flow of the stream. A heavy boom protects it against floating débris and there are grooves for stop logs

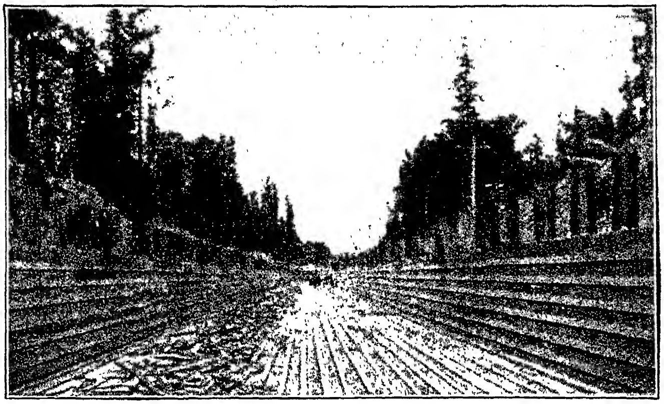


Fig. 207. Typical Timber-Lined Canal
Courtesy of "Engineering News"

and headgates worked by a gasoline engine, or by hand; a sluiceway has been built to discharge small boulders and gravel from the intake.

A wood flume, 28 feet wide, 8 feet deep, sloping 7 feet to a mile, carries water from the intake a mile to the first of the settling basins.

Settling Basins. A chain of settling basins about 2 miles long was made from four flat benches between a bluff on the south and a steep hillside on the north, which drops down toward the river. These benches are of practically uniform elevation and are separated by projecting points of land which have been cut by canals. Embankments along the northerly side completed the basins.

The first of the chain is long and narrow, admirably adapted to deposit all heavy silt before reaching its outlet. Sluice gates have

been provided so that the basins may be drained and the silt sluiced out; they have also been provided with generous spillways to avoid being flooded above the intended level.

From the last basin an unlined canal about 7000 feet long leads across a flat-top summit. The maximum cut is 35 feet and the canal is 40 feet wide at the bottom, sloping slightly under 1 foot to the mile. As it is cut through a hard-clay conglomerate and the section insures low velocity, there is no fear of erosion.

After crossing the summit this canal drops 111 feet in 2 miles to a natural valley, and here the canal has been timber lined, Fig. 207.

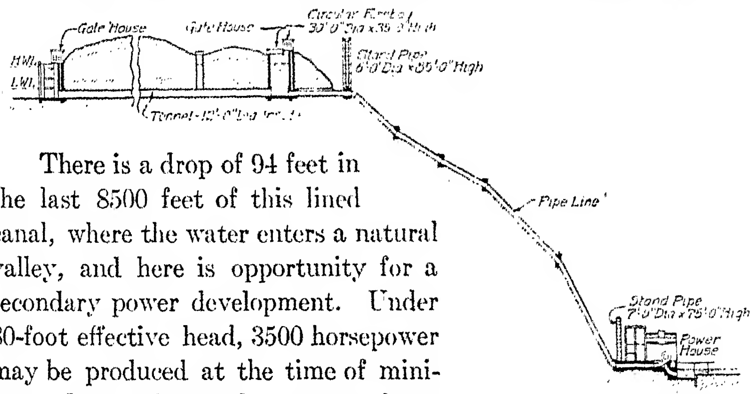


Fig. 208. Profile of Tunnel and Pipe Line, White River Development

There is a drop of 94 feet in the last 8500 feet of this lined canal, where the water enters a natural valley, and here is opportunity for a secondary power development. Under 80-foot effective head, 3500 horsepower may be produced at the time of minimum flow and 7000 horsepower for 9 months each year. The chain of settling basins could be used as storage for 7000 horsepower on daily peak loads continually through the year.

From the canal the water flows 3500 feet through this valley until it reaches another valley extending at right angles. The water has been led across the valley and is confined between two 1600-foot embankments far enough apart to give low velocity and additional storage. Then it passes 2900 feet through a wide cut on a slope of 5 feet to the mile and discharges into the new Lake Tapps near the old Lake Church, Fig. 208.

Storage and Conduction. The Lake Tapps storage basin includes Kirtley, Church, and Crawford lakes. Numerous small ravines lead from this natural basin to the White River Valley and these have been enclosed by embankments.

At the northwestern corner of old Lake Tapps is a long narrow indentation, at the northern end of which a cofferdam was built to

permit excavation of an outlet canal beyond. The cofferdam was made heavy and permanent, with provision for stop logs in case it was desired to unwater the outlet.

This canal was the heaviest cut on the work, 600,000 cubic yards being excavated in a length of 2500 feet; the greatest depth was 90 feet. A bottom width of 45 feet was kept throughout except at the western end, where 125 feet was needed for 300 feet to form a forebay in the front of the tunnel portal at the western ridge. The maximum capacity of the canal is 3000 cubic feet per second and a heavy rectangular covered timber conduit was laid in the bottom to prevent erosion.

The tunnel, Fig. 208, through the western ridge is a smooth straight circular conduit 2850 feet long, lined with reinforced concrete, and has an inside diameter of 12 feet and a 12-inch minimum thickness of lining. The entrance portal is of concrete with a bell mouth to reduce entrance head and to support the gate house and racks. The gate is electrically operated and has a by-pass; air shafts in the gate house prevent a vacuum when the gate is closed. The rack bars are of $\frac{3}{8}$- by 5-inch iron and form a straight panel 58 feet high by 40 feet wide. The rack is provided with motor-operated cleaning devices.

The western end terminates in a circular forebay 30 feet in diameter and 35 feet high, located 250 feet back from the western brow of the hill. The pipe line for each turbine starts at this forebay, smaller concrete-lined tunnels leading out to the brow of the hill. At the top of the forebay is a gate house and a spillway capable of discharging all possible surges here. The upper half of the pipe tunnels is oval-shaped and the lower half is circular, 9 feet in diameter. An 8-foot pipe is drawn into each tunnel and connected to an 8-foot gate valve concreted in the forebay lining. The oval tunnel section leaves room for inspection of the pipe.

Ultimately, there will be six pipe lines and tunnels, three connected with a similar forebay not yet built. At the outlet of the tunnels the steel pipe lines are connected with a header from which the small exciter pipe leads. The exciter line can thus be supplied from any of the main lines.

From the tunnel portals the pipe lines follow the hillside contour for some 2000 feet. The diameter of the pipe line varies from 8

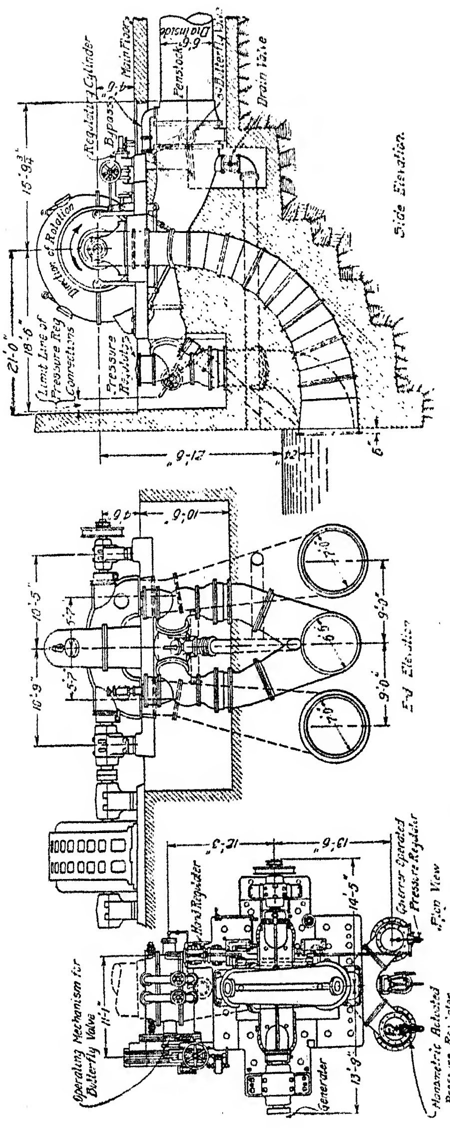


Fig. 209. Details of 18,000-H.P. Francis Turbine Installed for White River Plant by Allis-Chalmers Company
Courtesy of "Engineering News"

feet above to 6 feet below, and varies in thickness from $\frac{3}{8}$ inch to $\frac{15}{16}$ inch.

Tailrace. The tail-bay is shaped for easy entrance to the race, the latter extending 2000 feet west of the power house to the river. At the entrance to the race, provision is made for stop logs to form a weir for measuring wheel discharge. The race is excavated in low marsh and peat land and double rows of piling hold the banks. The section is wide for low velocity.

172. Operating Equipment. *Safety Provision.* Multiple provisions have been made against surges in the long column of water. There is first an open standpipe 6 feet in diameter, 85 feet high, at the upper end of each pipe line. The top of these is 25 feet above high water. At the lower end of each line, and outside the power house, are two air chambers 7 feet in diameter and 75 feet high, containing 2000 cubic feet of air under 190 pounds pressure maintained by the hydraulic head. The air cushions the water column for each unit independently and permits governing without waste. These cushion tanks are fitted with

floats and valves to allow free water passage in and out, but to prevent escape of air into the pipe line. As a third measure, each turbine casing is connected to two 30-inch relief valves, one operated by the governor and the other by pressure rise. Bursting plates are inserted in a separate discharge pipe, to give way in case the valves fail.

Units. The design and construction of the hydraulic equipment is described and illustrated in detail in *Engineering News,* April 18, 1912, and it may be sufficient here to say that the main turbines, Fig. 209, are a spiral-case double-discharge single-runner high-head Francis type normally rated at 18,000 horsepower, but capable of developing 20,400 horsepower under increased head. The wheel cases are of cast steel; the bearings are a water-cooled, oil-ring type; water-cooled friction brakes have been provided for quick stops. The guaranteed efficiency is 84½ per cent. Only two wheels have been installed, but the present building has room for two more; the ultimate installation of six will require an extension. The main generators are a 10,000-kilowatt 6600-volt three-phase type showing factory tests of 96.8 per cent efficiency. Two 720-horsepower impulse-wheel exciter units have been installed, with 225-kilowatt 240-volt direct-current generators.

There is a 550-cubic-foot motor-driven air compressor for charging the air-cushion tanks and for general station use. The central governor-oil system has motor-driven pumps.

Regulators. The governor adopted here is a type with the familiar fly-ball lift of a floating lever carrying the regulating-valve stem compensating relay and a motor attachment for switchboard control. The governor regulating valve is double acting and hydraulically balanced so that practically no energy is required to raise or lower the valve stem. The fly balls are extremely sensitive but still release considerable energy so that the smallest speed variation will move the control valve. A speed variation of $\frac{1}{5}$ of 1 per cent is sufficient to cause motion of the turbine gates. The regulating valve returns to its mid-position after each movement of the compensating relay acting on the other end of the floating lever.

To protect the pipe line and to assist the governor, serious surges in the water column must be prevented. Inertia in the water

column has been overcome in three ways at this plant: (1) There is a compressed-air cushion in chambers outside the power house connected to each penstock; (2) two 30-inch relief valves are connected to each turbine casing, one of these being opened by the governor to by-pass water when the speed gates are closed and closing thereafter gradually at a predetermined adjustable rate; (3) breaking plates which are provided to give way in case of failure of either of the other devices.

The equipment was turned over to the operating staff March 1, 1912, and is now regulating the whole power system exclusively.

EDISON ELECTRIC COMPANY

(Los Angeles)

173. Water Supply. The Mill Creek Power Plant No. 3 of this company at Los Angeles, California, which went into service in 1903, is remarkable for the high head used. All the water usually flowing in Mill Creek at Akers Narrows is diverted by a masonry dam, and conducted through 5 miles of pipe to a reservoir 1960 feet above the power house in Mill Creek Canyon. The conducting pipe slopes 0.2 foot per 100 feet, and is designed to carry 20 cubic feet of water per second. It contains 5 inverted siphons of steel pipe, aggregating 3585 feet in length, and 25,190 feet of concrete pipe 3 inches thick and 32 inches in inside diameter, and passes through 10 tunnels having an aggregate length of 7500 feet.

From the reservoir the water descends through a steel pressure pipe, varying in diameter from 26 inches to 24 inches, and in thickness from No. 14 Birmingham wire gage to $\frac{7}{8}$ inch. The pipe is protected from rust by a heavy coat of asphaltum, applied by dipping. At the lower end it branches, leading the water through 18-inch and 14-inch lap-welded pipe to the four generating units, which are housed in a concrete building with steel roof trusses and galvanized-iron roof. Of these generating units, three were made by the Abner Doble Company.

174. Power Units. Each Doble unit consists of a 1300-horsepower Doble tangential water wheel and a 750-kilowatt generator, mounted on a single shaft. This shaft has a speed of 430 revolutions per minute, and is mounted in three bearings which rest on a single cast-iron base frame set in concrete. Each wheel is provided with a

Doble needle-regulating and deflecting nozzle, with hand-operated balanced needle. With this apparatus the station attendant can set the needle by hand every half hour at the most economical point in order to carry the load which from experience he is led to expect during the next half hour. The governor takes care of all sudden fluctuations of load by deflecting the nozzle momentarily, so that all or part of the water issuing passes under the water wheel, and wastes its energy against the "vortex" baffle plate installed in the tailrace.

The static pressure due to the head of 1960 feet is over 850 pounds per square inch, and the spouting velocity of the jet is about 4 miles per minute. The generating units deliver current at 750 volts to the switchboard, whence it passes through transformers, and out over the 33,000-volt 86-mile transmission line to Los Angeles.

CENTRAL WEST DEVELOPMENT

MISSISSIPPI RIVER POWER COMPANY

175. General Characteristics. The work is a water-power development in the Mississippi River at the foot of the Des Moines Rapids, formed by the construction of a dam across the river between Keokuk, Iowa, and Hamilton, Illinois, with a power house on the Iowa end, flanked by a lock, a dry dock, and a sea wall protecting the new shore line and elevated railroad.

The plans provide for the ultimate installation of thirty units, each consisting of a turbine and electric generator direct-connected on the same shaft, with a capacity of 7500 kilowatts, or about 10,000 mechanical horsepower, each. After deducting reserve, operating loss, and other mechanical debits, there is offered for sale and uniform delivery 200,000 horsepower of electric current.

Natural Facilities. This development is made possible by the topography and geology at its site. The Des Moines Rapids represent a comparatively recent diverting of the Mississippi River, caused by a glacier damming its former channel farther to the west. The conditions which caused the river to cut a valley 5 to 10 miles wide in the preglacial age were absent in the postglacial age when the channel was cut through where are now the rapids.

The result is, that the bluffs at the rapids are closer to the present river than at any other point in its entire length, and are close to the banks most of the way from Keokuk to Montrose.

This allows a dam at the foot of the rapids with comparatively small area of overflow by the pool above; and this obviates the objection obtaining elsewhere, that the cost of the flood damages above the dam renders the development of many water powers commercially impossible. The bottom of the river here is hard blue limestone, affording an ideal footing for the dam and other works, from an engineering viewpoint. The slope of the rapids is steeper than in other stretches of the Mississippi, and the fall in the twelve miles above the dam is 23 feet.

176. Outline of Works. *Plant.* Dam, power house, lock, dry dock, sea wall, ice fender; all one concrete monolith with a total linear measurement of 13,185 feet, or $2\frac{1}{2}$ miles.

Dam. Length 4278 feet, plus east abutment 290 feet, and west abutment 81 feet, total 4649 feet, or 0.88 mile; width 29 feet at top and 42 feet at bottom; height of structure 53 feet; composed of 119 arched spans, with piers 6 feet thick and span 30 feet inside; spillway sections each 30 feet long and 32 feet high. Steel gates on top of spillways 11 feet by 32 feet. Upstream face vertical; downstream face of spillways an ogee curve. Dam keyed into limestone bottom of river about 5 feet.

Power House. Length 1718 feet, width 132 feet 10 inches; total height 177 feet 6 inches. Substructure of monolithic concrete; walls of reinforced concrete; trussed roof. Four floors; the first containing generators and accessories, and the others containing oil switches and electric accessories. Substructure 70 feet high to generator floor, or 78 feet high to transformer floor. Superstructure 107 feet 6 inches from generator floor to pinnacle.

Installation. Thirty units, Fig. 120, Part II, and Fig. 210 having turbine and generator on same vertical shaft—with four auxiliaries added; first installation 15 power units and 2 auxiliaries.

Turbine. Francis-type, special design; capacity 10,000 horsepower; overload capacity 13,500 horsepower, efficiency 88 per cent*; 57.7 revolutions per minute; diameter at buckets 16 feet 2 inches; thrust bearing lubricated by forced-pressure oil, and supplemented also by oil-immersed roller bearing—the single thrust bearing set above water carries rotating parts weighing 550,000 pounds. Tur-

*Later tests of the turbines of the construction illustrated in Fig. 120, Part II, showed an efficiency of 90.44 per cent.

bine is direct-connected with generator on hollow vertical shaft 25 inches in diameter. There are 16 buckets on the hub. Rating

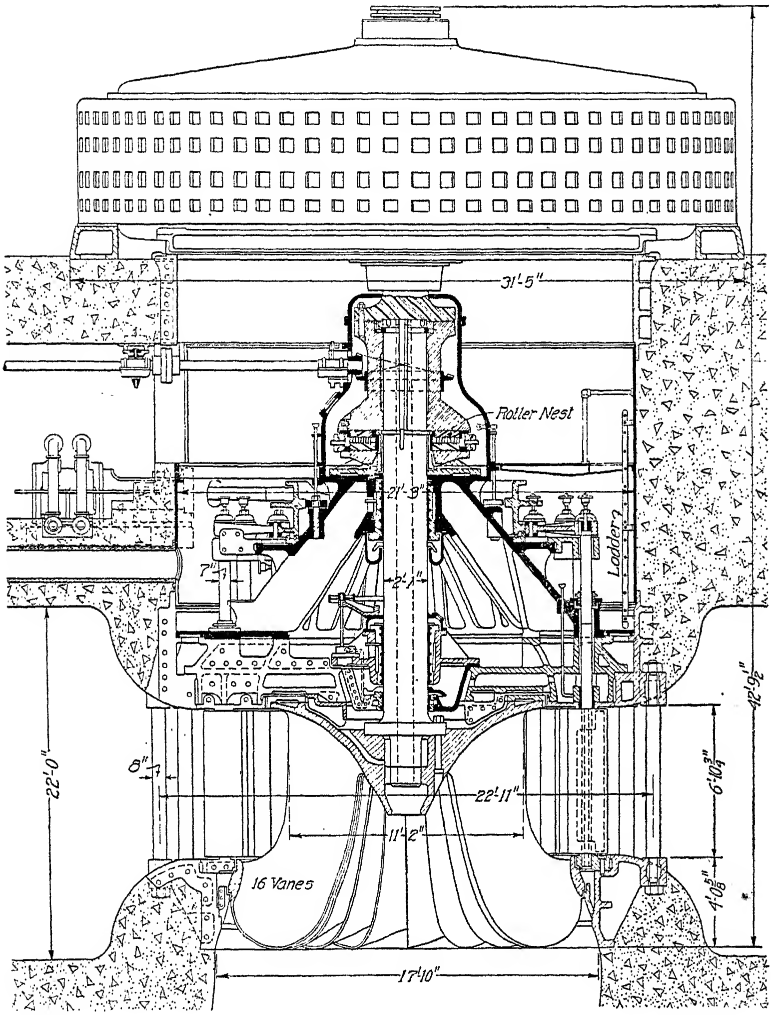


Fig. 210. Sectional Elevation of Francis Type Turbine for Mississippi River Power Company
Courtesy of I. P. Morris Company, Philadelphia

is based on head of 32 feet. Regulation is by guide vanes at inner circle of scroll chamber actuated through a system of levers by fly balls acting on a cylinder of compressed oil.

Hydraulics. For each unit there are 4 intakes converging into a scroll chamber 39 feet in diameter with spiral floor, the whole so shaped that the water strikes every point on the circumference of the turbine with equal velocity and impulse. Draft tubes below the turbines are vertical and 18 feet in diameter at the top; and at the bottom are horizontal, oblong shape, 22 feet 8 inches in vertical diameter and 40 feet 2 inches in horizontal diameter. Bottom of draft tubes and bottom of tailrace are set 25 feet below the bottom of the river. Tailrace is excavated parallel with east wall of power house and down to the railroad bridge.

Generator. Capacity 7500 kilowatts. Current alternating 25-cycle three-phase, generated at 11,000 volts and stepped up to 110,000 volts for transmission. Field revolves 57.7 revolutions per minute, and is excited by auxiliary units through motor-generator set for each one. Diameter of generator on the floor 31 feet 5 inches; diameter revolving field, 25 feet 5 inches; height at center 11 feet 3 inches.

Transmission Lines. One to St. Louis 144 miles; voltage 110,000; six copper cables $\frac{5}{8}$ inch in diameter, each composed of 19 wires; supported on steel towers set on concrete foundations in right of way 100 feet wide. One to Burlington 40 miles; voltage 11,000; aluminum cables on poles.

Lock. Width 110 feet; lift 40 feet; height of walls 52 feet 4 inches; thickness of walls 8 feet to 33 feet; length of lock 400 feet inside and 618 feet 6 inches outside; depth on sill 8 feet. Gates 115-foot span, steel trusses, pivots hardened steel on bronze; motive power pneumatic from water power. Time of lockage 10 to 15 minutes.

Dry Dock. Location between lock and Iowa shore. Dimensions 150 feet by 463 feet.

Sea Wall. Length 1110 feet; height 45 feet to 73 feet. Function to protect railroad tracks.

Ice Fender. Location from upper eastern corner of power house to Iowa shore; direction somewhat curved. Length 2325 feet of concrete construction and 300 feet of floating boom of timbers, with concrete abutment at Iowa shore. Concrete portion is composed of 29 spans with 10-foot piers and 60-foot openings; top of structure is 5 feet above high water, and the top of the open-

ings 4 feet below low water. Width 8 feet at top and 16 feet at bottom. Water will flow into forebay through openings between piers; ice and drift will be stopped by the top. Floating boom to be opened back against the shore during navigation season.

Ice will form on lake above dam in winter and, in the comparatively still water there, will lie as an ice field until it melts.

177. General Description.* This plant is remarkable not only for its unprecedented size but also for its many novel features of construction. The ultimate plant will contain thirty turbines, each having a rated capacity of 10,000 horsepower at the normal head of 32 feet. At the maximum head of 39 feet the output of each of these units will be approximately 14,000 horsepower and at the minimum operating head of 20 feet approximately 6000 horsepower. The power company has a large auxiliary steam plant available in St. Louis, on the switchboard of which a considerable part of the output of the Keokuk plant will be furnished. This arrangement will give them a commercially available primary load, even under the worst conditions, of more than 200,000 horsepower, and it is probable that a large secondary load will be developed in addition to this. Furthermore, it should be remembered that whereas the commercial output of primary power will not greatly exceed 200,000 horsepower, nevertheless the total installed capacity, based on normal rating is 300,000 horsepower, which places this plant in a class by itself. The excess installed capacity is necessary in order to provide for the low-head condition when the power of the turbines is materially reduced.

Arrangements of Units. The type of turbine and its setting in the power house are both novel in American practice. This is the first important plant designed for large slow-speed generators, direct-connected to single-runner vertical turbines installed in wheel chambers of spiral form built in the concrete foundations of the power house. The spiral casing is a common form in high-head turbine practice, but is always made of metal. The Keokuk casings are built on the same principle, but are formed in the concrete from wooden forms.

These turbines run at 57.7 revolutions per minute, which is the lowest speed thus far recorded in American practice for direct-

*Chester W. Larner, hydraulic engineer, *Engineering Record*, March 15, 1913.

connected units of large size. At the time this speed was selected, however, it was the maximum obtainable for a single runner under 32-foot head with high efficiency. The efficiency obtained at the Holyoke testing flume with a small-size model of this runner was over 88 per cent. A further restriction on the speed was imposed by the 20-foot head condition at which time it was vitally important to get as much power as possible at 57.7 revolutions per minute.

The speed of these units could have been increased by the use of two runners instead of one, as has been done frequently in the past, but the double-runner construction does not give the same degree of accessibility, nor the same compactness of installation, as the single runner. These defects more than offset the advantage due to the increased speed.

Each unit consists of a single runner with a set of balanced wicket gates or guide vanes having an operating mechanism of the exposed type under the control of a governor with its actuator on the generator floor. The great advantage of the single-runner type of unit over a multiple-runner type is clearly shown in the fact that all of the regulating mechanism and moving parts of the turbine, except the gates themselves and the runner, are outside of the water passages and accessible for adjustment and lubrication.

Turbine Gates. The turbine gates, which swing on pivots and control the quantity of water supplied to the wheel, are steel castings, each weighing more than a ton and being larger in dimensions than an ordinary door. Each gate has a hollow forged-steel stem passing through it, with bronze bearings in the crown and curb plates at the top and bottom of the gates. These stems are keyed to the gates, but may be readily withdrawn if it should be necessary at any time to remove any of the gates for replacement or repair. This is a novel feature in gate mechanisms of this type and is only possible on wheels of unusually large size. All bearings of the gate mechanism are grease-lubricated, the lower bearing of the gate stem being supplied by a hole bored through the entire length of the stem.

The operating ring is a heavy steel casting carried on a continuous ring of hardened-steel balls and guided circumferentially by forged-steel rollers. These rollers run on eccentric pins, making it possible to take up any looseness between the rollers and the ring, centering the latter accurately. The operating ring is connected by

two forged-steel rods to the crossheads of the regulating cylinders, operated by oil under 200-pound pressure. The supply of oil is controlled by the governor actuator placed on the generator floor.

Shaft and Bearings. The main shaft is 25 inches in diameter and about 21 feet long, with an 8-inch hole bored through it. It has a combination thrust block and coupling of cast steel connecting the turbine with the generator, Fig. 210.

There are two steady bearings in addition to the thrust bearing both of which are babbitted and lubricated by a gravity supply of oil. The thrust bearing used on twelve of the first fifteen units is a combination roller and oil-pressure bearing made by the Standard Roller Bearing Company, Philadelphia. It is arranged for an oil supply at 250-pound pressure and so designed that the oil pressure is sufficient to lift slightly the entire weight of the rotating parts of both turbine and generator, amounting to a total of about 550,000 pounds, thus entirely relieving the rollers of the load. While operating in this way, if the oil pressure should fail, the thrust block will settle on the rollers, transferring the load to them.

The other units will be equipped with thrust bearings of the Kingsbury type. This is a comparatively new form of thrust bearing which has shown very promising results under similar conditions. It is a babbitted bearing of novel design, where the circulating oil is under atmospheric pressure only. It requires no auxiliaries other than a gravity oil feed and has been successfully operated under loads as high as 900 pounds per square inch of bearing area. The coefficient of friction has been demonstrated to be lower than that of roller bearings under the same conditions. These bearings are interchangeable with the roller bearings, and either type may be used should it demonstrate any considerable superiority.

Governors. The governors for these wheels are as notable for their size as the wheels themselves. They were built by the Lombard Governor Company, Ashland, Massachusetts, and have a capacity of 250,000 foot-pounds each. In designing these governors it was found necessary to separate the parts of the control mechanism, installing the actuator on the generator floor and the two regulating cylinders or servo-motors, together with the relay valve, in the thrust-bearing gallery. Here also are placed the accumulator and receiver tanks and the large triplex pumps which supply the pressure.

DEVELOPMENTS WITH UNUSUAL FEATURES

INSTALLATION AT MILL CREEK, UTAH*

178. Supplementary Supply. In laying out hydroelectric systems it is sometimes possible to add appreciably to the output of an existing power station by introducing into the main flow line additional water from some source other than that of the primary supply. In many instances this can be accomplished with an expenditure of money which is small in proportion to the gain in power output of the station. In plants where the supply of power is equal to, or very little in excess of, the demand, the value of the additional capacity thus made available becomes very great, particularly during the low-water period.

Sometimes it is possible to gain the additional water by diverting some small stream or spring whose channel is at a sufficiently high elevation to allow its waters to be carried by gravity through a conduit to the main flow line. If, on the other hand, the source of the spring is at a lower elevation, it will be necessary to use a pump to lift the additional water to the main conduit. The latter method would be profitable only, of course, when the head against which the water must be pumped is small in comparison with the effective head at the plant where it is to be used.

An example of a profitable development illustrating the latter case is the Mill Creek pumping station herein described.

179. Features of Plant and Flow. The two plants of the system most intimately associated with the pumping plant to be described are those in Mill Creek Canyon, southeast of Salt Lake City, Utah. The lower plant, known as Mill Creek No. 2, is located at the mouth of the canyon, approximately 7 miles southeast of Salt Lake City. The upper station, Mill Creek No. 1, is situated about 5 miles farther up the canyon. The accompanying tabulation, which shows the amount of water in second-feet available each month at the lower plant, is characteristic and illustrates the nature of the flow.

The water passing through the upper plant is discharged into the intake reservoir for the lower plant, the diversion works for the latter station being about 650 feet down the canyon. A small

* From "Pumping Water to Increase the Supply for a Hydroelectric Plant", by M. B. Lott, *Engineering Record*, March 15, 1913.

Flow Available at Lower Mill Creek Plant

(Monthly Variation during 1912)

Period	Rate		
	Maximum (sec.-ft.)	Minimum (sec.-ft.)	Average (sec.-ft.)
January	9.36	8.88	9.16
February	9.10	8.45	8.82
March	10.56	8.85	9.30
April	16.44	12.43	14.62
May	51.84	16.61	30.91
June	56.00	24.50	37.00
July	18.99	12.61	16.38
August	17.67	13.47	15.26
September	13.59	12.68	13.11
October	12.57	10.73	12.01
November	12.16	11.09	11.56
December	11.50	10.07	10.90

concrete dam, approximately 20 feet high, backs the water up to the upper plant, this forming a small reservoir with a storage of approximately 6 acre-feet, an amount of water equivalent to about 560 kilowatts at the lower plant for 8 hours.

The intake to the pipe line is located at a point about 19 feet below the crest of the spillway, and the pipe line itself leading to the lower plant is laid on a grade of 2 feet per 1000 feet.

The static head is 1031 feet. The accompanying curve, Fig. 211, shows the net effective heat at the power house for any discharge between zero and the maximum discharge capacity of the conduit—19.3 cubic feet per second.

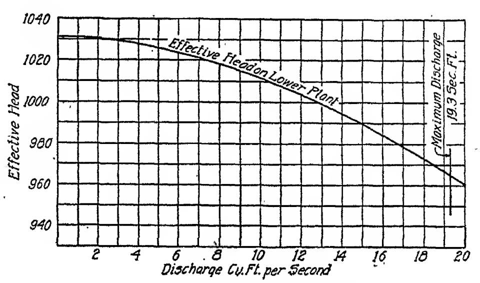


Fig. 211. Relation of Discharge to Effective Head in Mill Creek Installation

There are two units at the lower plant, each consisting of a Doble water wheel direct-connected to a General Electric generator. Each water wheel is capable of developing 1300 horsepower at 514 revolutions per minute under an effective head of 990 feet.

The generators are rated at 700 kilovoltamperes, 60 cycles, and 2300 volts. One of the curves, Fig. 212, shows the kilowatt output at 100 per cent power factor for different quantities of water discharged through the wheels. This is a theoretical curve which was checked by actual test, using a water rheostat for load. The lower Mill Creek plant does most of the governing for the system.

Springs and Pumping Station. Approximately ½ mile down the canyon from the intake works for the lower plant a number of springs issue from the ground within a limited area. The flow from these springs, by weir measurement, varies from 3 second-feet to 4.5 second-feet, depending on the season of the year. By means of a small diversion dam and pumping plant it was possible to collect these waters and pump them into the main 30-inch flow line, thus increasing the available water supply at the lower plant and in consequence its available power output.

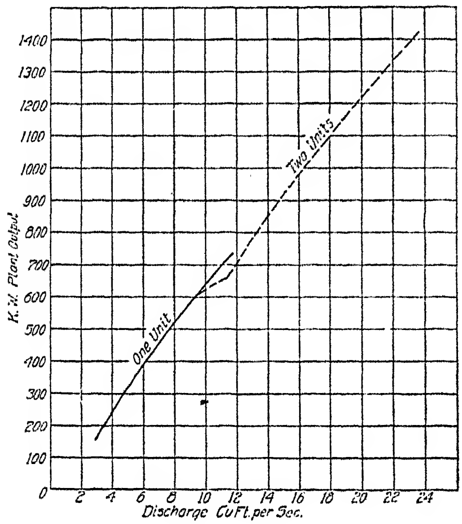


Fig. 212. Curve Showing Output at 100 per cent Power Factor at Mill Creek Installation

The water flows from the stream channel into a receiving well 6 feet by 12 feet and 7 feet 2 inches in height. This is topped with a reinforced-concrete cover on which the motor-starting apparatus is placed.

The horizontal pumping unit consists of a Worthington single-stage double-suction 6-inch centrifugal pump, direct-connected to a General Electric three-phase squirrel-cage induction motor, rated 100 horsepower at 1800 r.p.m., 2200 volts, and 60 cycles. The energy for driving the motor is supplied from the upper Mill Creek plant. The pump is designed to lift 3.5 cubic feet of water per second in ordinary operation, the normal head against which the water must be pumped being 138 feet, including friction. In Fig. 213 is given the performance of the plant.

A 12-inch wire-wound wooden stavepipe carries the water pumped to the main 30-inch flow line. A gate valve is provided in the 12-inch pipe near the point of connection, while a vent pipe is connected in just below, so that the pumping-plant line may be emptied when desired.

Power Value of Additional Water. From the table of water available at the lower plant (other than that pumped), the output curve of the lower plant, and the performance curve of the pumping unit, it is possible to estimate the value of the pumping installation. From the water-supply table it will be seen that during the months of May and June the water is in excess of the pipe-line capacity, so the use of the pumping plant is out of the question during those two months, but the spring water is available for use during the remaining portion. From the output curves for the two plants it will be seen that the 3.5 second-feet will, on the average, be good for 237 kilowatts at the lower plant, with a motor input of 67 kilowatts, making a net gain of 170 kilowatts. On the basis of having 3.5 second-feet available during ten months, the total possible net yearly gain in energy resulting from the installation of the pumping plant will therefore be 1,224,000 kilowatt-hours. Estimating that one-half of this could be sold at 1 cent per kilowatt-hour, the saving would amount to $6120. This is more than the initial cost of the pumping system and shows that the return on the investment can be large.

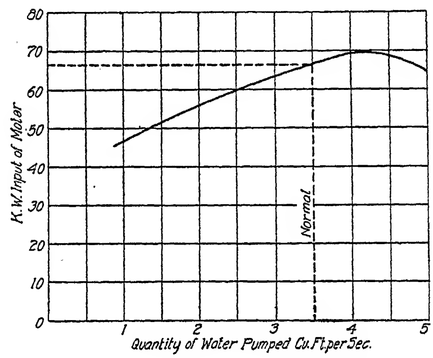


Fig. 213. Performance Curve for Pumping Plant for Mill Creek Installation

Not only is the water supplied by the springs good for immediate use at the lower plant, but at times when it is not needed at the lower plant it can be pumped into the intake reservoir for storage until time of peak load.

The total cost of the pumping-plant system was approximately $32 per net kilowatt of rating gained.

PIKE'S PEAK HYDROELECTRIC COMPANY

180. High Head. This company, of Colorado Springs, Colorado, has the distinction of operating a water-wheel plant under the highest head available in the United States. In fact, there is but one installation in the world utilizing a higher head, and that for only a small amount of power—namely, the Vauvry plant in Switzerland, subsequently described.

The plant in question is located on the outskirts of the town of Manitou, Colorado, and consists of three Pelton units, each direct-connected to a 750-kilowatt electric generator running at 450 revolutions per minute. The net head on the Pelton wheels is 2150 feet, equivalent to the enormous pressure of 935 pounds per square inch.

The wheels are mounted in the pulley compartment of the generator, and are provided with combination needle and deflecting nozzles operated by hydraulic governors. The gates, nozzles, and other pressure parts are of cast steel, designed with a large safety factor, and were subjected to a cold-water test of 2000 pounds per square inch before installing. The wheels proper consist of cast-steel disks with gun-metal buckets, fine ground and machined inside. Each wheel has an overload capacity of 1500 horsepower.

Current is transmitted to Colorado Springs for power and lighting purposes, and is also largely consumed by the many mines and mills in that vicinity.

PLANT AT VAUVRY, SWITZERLAND*

181. World's Highest Head. This installation is situated on the left bank of the river Rhone, a short distance above the point where that stream empties into Lake Leman. The water is taken from Lake Tanay, at an elevation of 4644.5 feet, and is delivered to the wheels in the power plant at an elevation of 1528.8 feet, which represents a gross head of 3115.7 feet—probably greater than that of any other hydraulic plant in the world. The installation is intended to supply electricity for lighting and power purposes to a large number of Swiss towns and villages in the valley of the Rhone. Lake Tanay, the source of the water supply, has an area of about 112 acres, and receives the drainage of an area of about 1875 acres,

*_Engineering News_, November, 1902.

which, it was estimated, would yield a steady stream of about 12.2 cubic feet per second throughout the year.

182. Structural Features. The main features of this plant comprise the supply pipe line and power house with its machinery.

Pipe Line. The head of the pipe line is at an elevation of 4559 feet, which is 65.6 feet below the normal level of the lake, and 84.2 feet below its maximum level; and it terminates in a vertical shaft about 41.7 feet deep. From the shaft a short gallery or tunnel 984 feet long, with a sectional area of 10.75 square feet, is built on an almost level grade. This tunnel is provided with bulkheads, pipes, valves, and other apparatus for regulating and controlling the supply of water taken from the lake; and at its end the pressure pipe line begins. For 328 feet, this line is a steel pipe 2.62 feet in diameter; then, for 984 feet, it consists of a masonry tunnel; and finally, for 3936 feet, it is again a steel pipe, 2.62 feet in diameter. At the end of the last section, the pipe branches into three pipes, each of 1.64 feet diameter. One of these pipes extends to the power plant; but the others are plugged, and will be built to the power house only when the demand for power necessitates their construction. At the point of junction of the single and triple pipes, the head of water is only 689 feet; but from this point on, the descent is very abrupt. At this point, also, there is a retaining valve, and a vertical standpipe (regulator) 1.31 feet in diameter, and 82 feet high, which relieves the water hammer in the pipe line above.

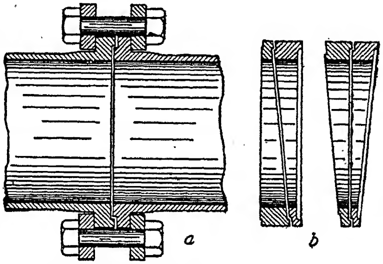


Fig. 214. Detail of Joint Connections for Pipe Line of Water-Power Plant at Vauvry, Switzerland

The steep-grade pipe line, from the junction point just mentioned to the power house, is 6363 feet long, and has a fall of 2952 feet. As stated, the pipe is 1.64 feet in diameter at the junction point, and it continues with this diameter for 2682.8 feet, varying in thickness of shell from about .275 inch to .45 inch, finally terminating in a **Y**, each branch of which is provided with a valve. From the branch of the **Y**, two 1.12-foot pipes extend for a distance of 4264 feet, with shells varying in thickness from about .3 inch to .7 inch. The transverse

joints are made as shown at *a* in Fig. 214; and in order that the joints may be tight, a suitable gasket is inserted before tightening up the bolts. As the pipe line lies in a trench following the surface grades, it has many bends; and to provide for these, the wedge-shaped pieces shown at *b* in Fig. 214 are inserted at the joints. The sections of pipe used varied in length from 16.4 feet to 32.8 feet, and weighed from about 1760 pounds to about 2500 pounds. Each branch of the pipe line has at its lower end a slide valve provided with a by-pass, to permit it to be operated by hand.

Power House. The power house is a steel building 45.9 by 216.5 feet in plan. The two pipe lines described terminate underneath its

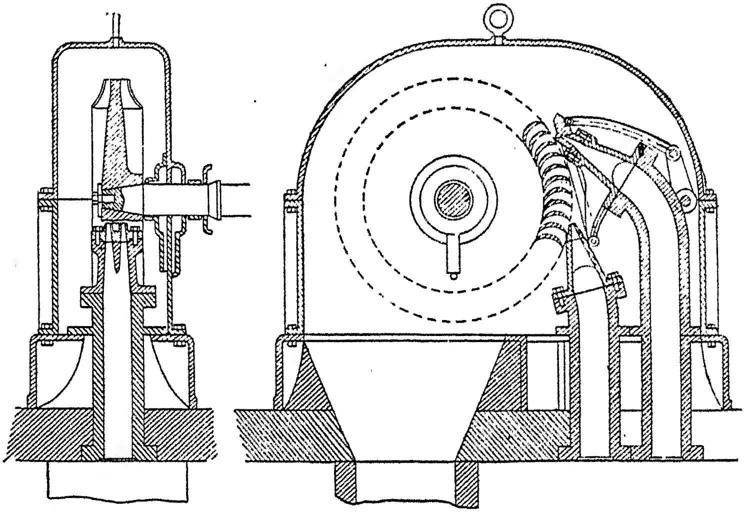


Fig. 215. Detail of Duvillard Wheel for Water-Power Plant, Vauvry, Switzerland

main floor; and each supplies water to two wheels, all of which are of the impulse type. Two makes of wheels were installed, as shown in Figs. 215 and 216. Each wheel is supplied with water through two nozzles, one above the other, the upper one of which is provided with a device actuated by a governor, for controlling the supply of water. A vertical diaphragm divides the end of each nozzle into two openings, as shown. The pipe from which the nozzles are supplied rises vertically from the main, and is opened and closed by a hand valve, above which it divides into two branches, one leading to each nozzle, and each opened and closed by a hydraulic valve controlled from the operator's platform on the floor above.

Each wheel is mounted on the alternator shaft, and operates at 1000 r. p. m. The dynamos are each of 500 horsepower.

SCHAFFHAUSEN INSTALLATION, SWITZERLAND*

183. Low Head. The city of Schaffhausen, situated some 2 miles above the falls of the Rhine, is supplied locally with electric power by low-head plants located on the river opposite the city, and having a maximum total capacity of some 4500 kilowatts. A low dam above the rapids flowing past the city gives a head varying with the state of the river from 12 to 15 feet. The power developed is all for local consumption, and is used for lighting, traction, and industrial purposes. The character of the load curve is quite similar to those with which American engineers are familiar, showing a pronounced peak in the late afternoon and early evening, especially in the short-day months of the year.

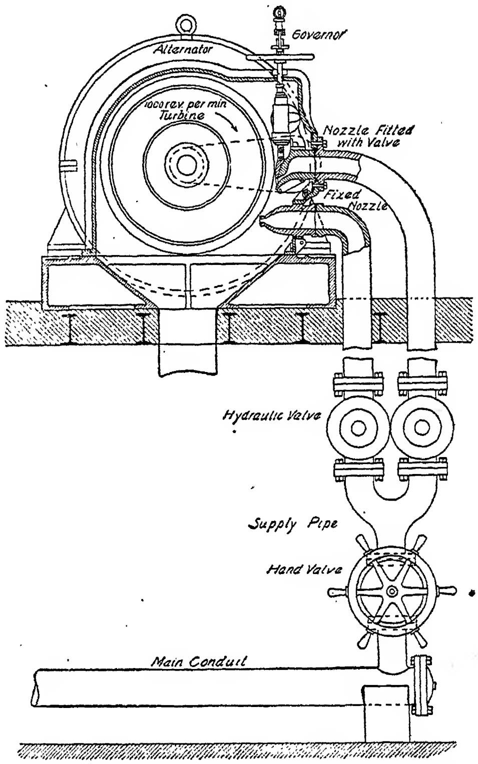


Fig. 216. Detail of Wheels Supplied by the *Societe de Constructions Mecaniques de Vevey*, for Water-Power Plant, Vauvry, Switzerland

184. Provision for Peak Load. For the purpose of carrying this peak without installing additional main units, there has been developed an interesting application of step-up transformation and storage. This consists essentially of the following elements:

(a) On one drive shaft with suitable connecting and disconnecting clutches three machines are installed: (1) a high-pressure

* Prof. W. F. Durand, *Engineering Record*, November 12, 1910.

multistage centrifugal pump; (2) an electric unit which, as desired, may be made to function either as motor or generator; (3) a high-pressure reaction turbine of the Francis type of about 1000-horse-power capacity.

(b) A pipe line with suitable valves connecting at will with either the pump or the turbine.

(c) A reservoir at an elevation of some 500 feet above the plant level.

At times when the capacity of the main plant units is in excess of the power demand, electric current is drawn from the mains and fed to the electric unit mentioned above, which, functioning as a motor, is connected to the pump and disconnected from the turbine. This results in taking water from the river and storing it in the elevated reservoir. Or, in other words, the energy of a relatively large amount of water falling through low head (with allowances for the necessary losses in the process) is transformed into the energy of a relatively small amount of water stored at high head. This is hydraulic step-up transformation and storage.

The result of this phase of the process is to maintain the elevated storage reservoir with always sufficient water to answer any sudden demand, or for regular daily peak service.

To utilize this storage of energy the connections are so changed that the pipe line is shut off from the pump and connected to the turbine, while the electric unit is disconnected from the pump and likewise connected to the turbine. The water being admitted to the latter, the combination functions as an ordinary turbine-driven generator delivering electric current into the line and thus aiding to carry the peak load. The entire scheme is simply a variant in detail of the application of well-known principles which have found application in all fields of engineering practice.

COST OF WATER POWER*

FACTORS OF VARIATION

185. Installation and Operation. Water power is generally, though not necessarily, the cheapest form of power development. The water itself costs nothing, but it cannot be used as Nature

*Articles 185 and 186 are abstracted from a paper by Professor Charles E. Lucke of Columbia University, New York, New York, on "Power Costs".

provides it. The cost of preparing it for proper use in wheels, and of providing for its control, will involve considerable expense, as a rule; and the interest on this expense will be correspondingly the largest charge against power cost, and will, so far as its importance is concerned, be commensurate with the fuel charges in gas, steam, or oil engines.

The cost of developing water power depends almost entirely on the local situation, and cannot be reduced to any formula or rule. In the early days of water-power development—say over 1000 horsepower—only low heads were utilized, and the wheels were located at the side of the dam, making the total development involve little more than a timber and stone dam, with a house at one end. As the demand grew for more power from the same stream, it was necessary to construct a canal for the purpose of bringing the water around the original power house to some other house. Thus the expense began to increase materially, from the difficulty of bringing the greatest quantity of water to the wheels under the largest heads by long pipe lines, or combinations of pipe lines, canals, flumes, tunnels, and vertical shafts.

The simpler development, complete, without electrical equipment, cost about $40 per horsepower. This has now increased so that a minimum of $75 per kilowatt is considered a very good proposition; while a maximum of $200 per kilowatt is not by any means prohibitive, including electrical equipment. The mean is somewhere near $100 for very large installations favorably placed, like those at Niagara Falls. Taking these two limits, and a gross interest charge of 5 per cent, with an average depreciation of 4 per cent, and with insurance and taxes at 1 per cent, there is a total charge of 10 per cent on the first cost, or $7.50 per kilowatt-year, as a low limit, and $20 per kilowatt-year as a high limit. The operating expenses, labor, oil, waste, repairs, etc., may be expected to cost from $1 to $5 per kilowatt-year, which places the cost of electric power at the busbars between the rare low limit of $8.50 per kilowatt-year, and the high limit of $25 per kilowatt-year for full load, 24-hour power, for these rates of charging expenses.

Besides the increase in development cost, there is another item that sometimes enters; and that is an increase in land expense or land damage, which may be large in settled communities. The

increase in cost is not all due to increase in cost of machinery; this has probably decreased, not only from better methods of manufacture and more competition, but also from the use of higher heads and better wheels and electric generators.

186. Competition. Just how much can be paid profitably for the development of water power, either now or in the future, will be measured solely by the power cost of that one of the competing systems—steam, gas, or oil—most available in the same locality, or, if not in the same locality, at some point within the limits of electrical transmission. To the cost of generating water power, must be added transmission cost, involving fixed charges on lines, transformers, switchboard, and other equipment, together with their maintenance and operating charges. All of these together may add to first cost \$30 per kilowatt, and increase the power cost for a 150-mile transmission \$5 per kilowatt-year.

After analyzing the details of power cost for oil, gas, and steam development, Professor Lucke continues:

These costs may now be summarized for comparison, as follows:

COMPARISON OF POWER COSTS

Assume: Stations consisting of six units, two in reserve, and four working on 24 hours rated load, with the exception of the water power. First cost and fixed charges based on capacity of 150 per cent of output.

	Water Power	Oil Engines	Gas Engine and Producer	Steam Plant
First cost per kw. rating	\$75.00–\$200.00	(160-kw. units) \$217.00	(600-kw. units) \$270.00	(5000-kw. units) \$110.00–\$150.00
Fixed charges, rate per cent	10 per cent	10 per cent	10 per cent	10 per cent
Fixed charges, per kw.-year	\$7.50–\$20.00	\$21.70	\$27.00	\$16.50–\$22.50
Operating and manufacturing costs per kw.-year	\$1.00–\$5.00	\$56.94	\$38.54	\$52.50
Total power costs per kw.-year	\$8.50–\$25.00	\$78.64	\$65.54	\$69.00–\$75.00

From these figures it appears that we have not yet reached the limit of cost of development of water powers which may be advis-

able. It apparently would pay to spend even more money than $200 per kilowatt, the present maximum for water-power development, if there were no other considerations entering. Among the chief considerations of this kind, may be set down that of transportation of products from the works, and of raw material to the works; but this must be considered against the question of transmission of current from the waterfall to a convenient point of transportation.

187. Load. The discussion and the figures presented above refer to a 24-hour continuous full load, a condition which frequently is practically fulfilled in the case of electrochemical works, but rarely in other branches of industry.

Load Factor. The term *load factor* may be defined as the ratio of the actual power output of an installation during a 24-hour period of operation to its 24-hour capacity at full load; or, in the case of the consumer, it may be defined as the ratio of average to maximum load. The term, however, is occasionally employed with a somewhat different significance.

Peak Load. The *peak load* represents the maximum output, corresponding to the maximum demand for power from all the industries served at any one time during a 24-hour period of operation; or, in the case of the consumer, it represents the maximum use of power during that period. So short a time as a 1- or 2-minute interval may be used in estimating peak loads.

The value of the load factor varies between very wide limits in the different industries, depending on their nature and peculiarities. Thus, from the census report of the year 1902, the average load factor of all United States lighting systems was about 23 per cent, while the street railways had a load factor of about 30 per cent, and an average of all electric-light and railway systems amounted to about 26 per cent. From these low values the load factor varies, for the different industries or combinations of industries, up to nearly 100 per cent, as stated above.

It is evident that in the case of an installation furnishing power to an industry using that power fairly uniformly, the load factor will be higher than under the reverse conditions. It is also clear that if the power is furnished to a combination of different industries so selected regarding power demand that the times of high requirement in some occur simultaneously with the times of low

requirement in others, the result of this correspondence will be a more or less uniform demand for power from the power company, and the load factor will be high.

Overload Compensation. The general effect of the load factor on the cost of power production is evident, for the plant must be designed and installed to meet peak-load conditions, even though these conditions obtain for short periods only during each 24 hours. Accordingly, during the balance of the time, part of the plant must lie idle or work at less than full load, whereas certain fixed charges on the total investment, such as interest, taxes, depreciation, and insurance, continue uniformly. Unless the entire plant representing the investment is put to its normal maximum use every hour in the day and every day in the year, it is not working at its theoretic maximum efficiency, and the cost of power production will be correspondingly increased.

To carry over the peak load, the motor units may be designed with sufficient overload capacity; storage batteries of sufficient capacity may be installed, to take the excess over the average load; auxiliary steam or gas units may be employed, or one or more main units may be held in reserve, for the same purpose.

COST TO CONSUMER*

AFFECTING CONDITIONS

188. Basis of Charge. "Since the day when the first commercial electric light entered the field of artificial illuminants, there have been endeavors to find an equitable way of charging for energy supplied in the form of electricity. At first, in the absence of any measuring instruments, the flat rate was the only method. This was soon found to be impracticable for most cases, and the ampere-hour meter, followed by various types of integrating and recording wattmeters, soon brought into use the idea of paying for the exact amount of energy used, at a given price per ampere-hour or per kilowatt-hour. This method is still in very general use in its simplest form, but there has been dissatisfaction with it from the time it started. The fact of the matter is that neither the straight flat rate nor the straight kilowatt-hour rate is equitable, except when applied in connection with a definite load factor; and even then it

* *Street Railway Journal*, June 30 and July 7, 1906.

may not be entirely so, due to uncertainty as to the number of hours per day that full-load conditions prevail, with corresponding high efficiency, and to the hours during which operation continues at light loads, with resultant low efficiency.

189. Load Factor. "It is fully recognized now, however, that the load factor is the root of the trouble; and unless a system of charging gives due consideration to it, there will always be inequality of rates and dissatisfaction on the part of the power company or of its customers, or of both. This has been shown in all classes of service—incandescent and arc lighting, heating and power purposes, including railway lines—and in power companies and consumers of all sizes."

The ratio of the use actually obtained to the theoretical or possible maximum use, is the load factor of the manufacturing establishment and of the railway line, just as it is of the power house or of the transmission system.

"The effect of load factor on cost of power is thoroughly understood where steam plants are concerned; but it might be supposed, in the case of hydraulic power, where no furnaces have to be banked, and inefficiency at light loads becomes unimportant, that the conditions would be different. Hydraulic turbines of modern design, however, usually have such characteristics that their overload capacity is very slight; and it therefore becomes necessary, if peak loads are to be handled, to provide extra machinery to take care of these. With no provision for peaks, it is still necessary to hold at least one generating unit in reserve, and a margin of capacity must be left unused in the operating turbines for gate travel in regulation, and to allow for partial clogging of distributors by refuse which accidentally enters the penstocks. As the water is available and costs no more if used to the full capacity of the plant, it is plain that the power-selling company will strive vigorously for a uniform load as high as is practicable for the installed machinery to carry. This results in making peaks a prohibitive element to power deals where the hydraulic plant has been some time in the field and has been able to discriminate in the choice of its customers.

"The plants now operating at Niagara Falls have been particularly fortunate in this respect, one of the oldest having a 24-hour load line of about 26,000 horsepower, and fluctuations not exceed-

ing 5 per cent of the average load. Needless to say, the portion of this power supplied for railway and lighting purposes is very small. Niagara conditions are unique, as the electro chemical plants, which consume the greater part of the present power, provide an ideal load.

190. Auxiliary Power. "The typical street-railway load necessarily has prominent peaks; and if these cannot be smoothed down by adjustments of service, it is still possible, where a fair price is asked for the water power, to carry the heaviest part of the all-day load by means of this, and the remainder by steam engines, gas engines, or storage batteries, or combinations of engines and batteries.

"The point is frequently raised that power companies undertaking to supply customers of any sort should be equipped to take care of all requirements of these customers, including peak loads. This is done in some cases, the power companies going so far as to provide steam plants for reserves and peak purposes. The character of local demands for power will usually determine this matter; and if the power companies eventually install auxiliary steam plants, it will be only because they are forced to it by periodic shortages of water, or inability to obtain customers whose aggregate use of power results in a high yearly load factor. The power company wants to sell all of its power all of the time; and in a thriving, progressive community, it is probable that it finally will come very near doing this. The load factor will improve as customers increase in number; and as the load approaches the full capacity of the plant, the power company will become more discriminating about closing new contracts, or renewing old ones, that involve conditions tending toward poor load factor.

"If power companies cannot entertain peak propositions at all, or if they place prohibitive rates thereon, the purchaser must then provide the steam plant, or storage battery, or both, to care for a part of the load. Railway systems supplied with purchased hydroelectric power afford ideal opportunities for the application of storage batteries. The batteries can be charged at night with power that otherwise could not be used; and the discharge of the load peak provides power at an extremely low load factor which costs only the fixed charges, operation, and maintenance of the battery.

191. Penalty. "If it is possible to make contracts for full power requirements, it is usual for power companies to place some

penalty rate on the peak power, or to arrange the terms of charge so that there are distinct advantages to the purchaser in keeping the load line as nearly straight as possible. The most common method is to sell a solid block of 'firm' power which can be used at a load factor of 70 per cent to 80 per cent, or better, charging the minimum flat rate for this, and providing power above the firm amount on a kilowatt-hour basis, at rates gradually increasing with the height of the peaks.

"Sometimes provision is made for charging extreme rates for possible peaks of such height that the railway company has no expectation of ever reaching them. These clauses should be avoided, if possible, as the unexpected is constantly happening in the operation and growth of a railroad. Where measurement of peaks is dealt with at all, it should be specified that they are not to be counted unless they continue for 2 minutes or longer. (In some cases 1 minute is specified.) Uncontrollable occurrences, such as the partial grounding of a feeder, or the performance of a defective car, may produce peaks of short duration which are of small consequence to the power company, but might be very costly to the railway company under an unreasonable power agreement.

"The purchaser should be allowed, without charge, swings of about 10 per cent (or 5 per cent) above the firm line of purchased power, provided the kilowatt-hours used above the line do not exceed those unused below it; since it is impossible to always carry the load directly on the limiting line, even with the aid of batteries and the most approved regulating devices.

192. Sliding Rate. "A fair method of charging for power is on a sliding rate depending on the monthly load factor. The maximum 2-minute (or 1-minute) peaks are recorded in kilowatts each day, and averaged for the month, giving the average maximum demand for the month. The total number of kilowatt-hours used during the month, divided by the number of hours in a month, gives the average hourly rate of consumption for the month. Then the monthly load factor is obtained by dividing this average by the maximum demand; and this factor is used as follows in determining the charge for the month:

Illustrative Examples. "Assuming that a manufacturer has made a contract to buy 400 horsepower for the operation of his

factory, and that the rate per horsepower-year varies between the limits of \$16 and \$43, depending on the load factor, the determination of his rate per horsepower per year for any given month would be as follows:

"If the kilowatt-hours consumed during a thirty-day month are 43,200, then the average demand for power is 43,200 divided by 720 (the number of hours in the month), equal to 60 kilowatts or 80 horsepower. Assuming further that his maximum demand each day was just 400 horsepower, then, of course, his average maximum demand for the month will be the same amount, and the load factor is 80 divided by $400 = .2$, or, as commonly expressed, 20 per cent. If the rate per horsepower-year varies between \$16 and \$43, it will be evident that the variable quantity is their difference, or \$27. The *rate* is therefore equal to the minimum rate (\$16), plus the load factor (.2) multiplied by the variable (\$27). In the present case, this will amount to $16 + .2 \times 27 = \$21.40$. The total charge for the month would therefore be $\$21.40 \times 400 \div 12 = \$713.33$, which is equivalent to 1.65 cents per kilowatt-hour, or \$107.00 per horsepower,* for power actually used.

"If the monthly load factor had been 30 per cent (.3) instead of 20 per cent (.2), the rate per horsepower per year would have increased to \$24.10; but the equivalent cost per kilowatt-hour would have decreased to 1.24 cents, a reduction of almost 25 per cent in cost per kilowatt-hour due to increasing the load factor to 30 per cent.

"This may readily be put in the form of an equation in which desired rate per horsepower per year is R, minimum rate limit is A, maximum rate limit is B, and load factor is L; thus

$$R = A + L\,(B - A)$$

"This method is much more equitable than that sometimes used, of selling all the power on a kilowatt-hour basis with a guarantee from the consumer of a specified load factor." With this system of charging, the method of establishing equitably the limiting values for horsepower per year (corresponding to \$16 and \$43, in the example above) will require very careful consideration.

193. Proportioning the Load. "Very careful consideration must be given to proportioning the division of load between water

*Cost per kilowatt-hour = cost per horsepower-year ÷ 6480.

power and steam power. The cost of hydroelectric power at 100 per cent load factor should be somewhere in the neighborhood of one-third the cost of steam-generated power at 100 per cent load factor, assuming reasonable first cost of plant and moderate distance of transmission in the first case, and average cost of coal and labor in the second. Obviously the bulk of the load should be carried by the purchased power; but the higher the limiting firm line of this power is raised, the lower will the load factors of both steam power and purchased power become, and the cost per kilowatt-hour of each will increase. In each case, however, there is a certain critical point to which the firm purchased-power line may be raised before the total cost (which is of prime importance) of combined purchased power and steam power will commence to increase. In raising the firm line of purchased power to this point, the total cost will be decreasing."

LOAD CURVE INDICATIONS

194. Loads of a Niagara Company. "Curve sheet, Fig. 217, shows at AA the remarkably straight local load line of one of the

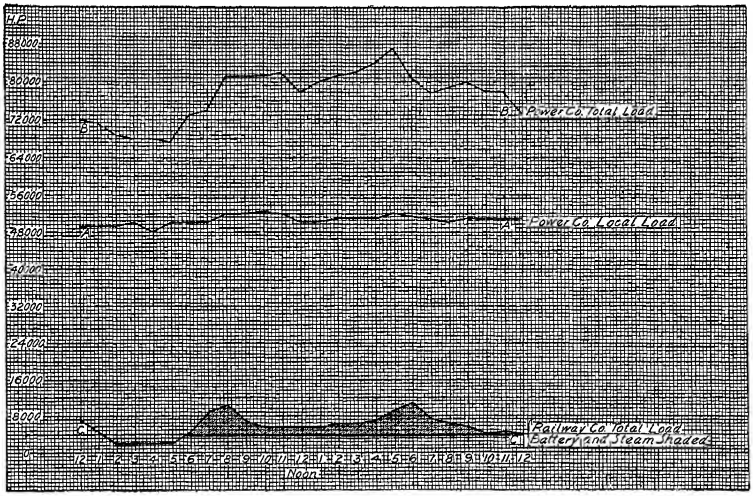


Fig. 217. Curve Sheet Showing Load Curves of One of the Niagara Companies

Niagara power companies. At BB is shown the total load line, including the long-distance load, of the same company. CC shows a railway load, the shaded portion of which is carried by the railway

company's steam engines and storage batteries. The unevenness of the power company's total load is not contributed to by the railway company, except to the extent of a dip during the early morning hours. The peaks of the railway load would, if included

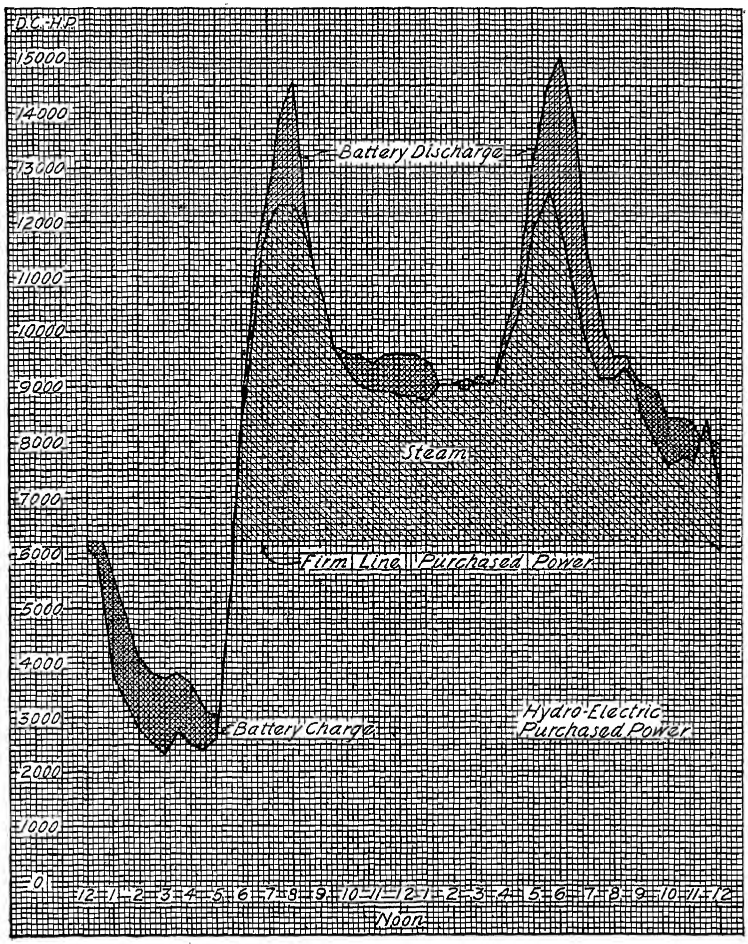


Fig. 218. Curve Sheet Showing Railway Load Line Represented as *C-C* on Curve Sheet, Fig. 217

in the power company total load, distort it considerably, in an undesirable way. The curves are all plotted from the same base line, and represent the same day."

Railway Load. "Curve sheet, Fig. 218, shows on a more open vertical scale the same railway load that is represented at *CC* in

217. The firm line of purchased power is here located lower, reference to the total load, than has been described as the economical point. This is partly for the reason that the chart represents a winter day (the heavy load season of the year). The total load drops below the firm purchased power line during the middle of the day at some seasons, and, as the firm line cannot be shifted back and forth, there are necessarily times when the proportions of purchased power and steam are not the most economical, as in the instance of this particular day.

Relation of Load Factor to Rates. "Fig. 219 gives the rates per horsepower in terms of load factor. The curve marked 'Hydro-

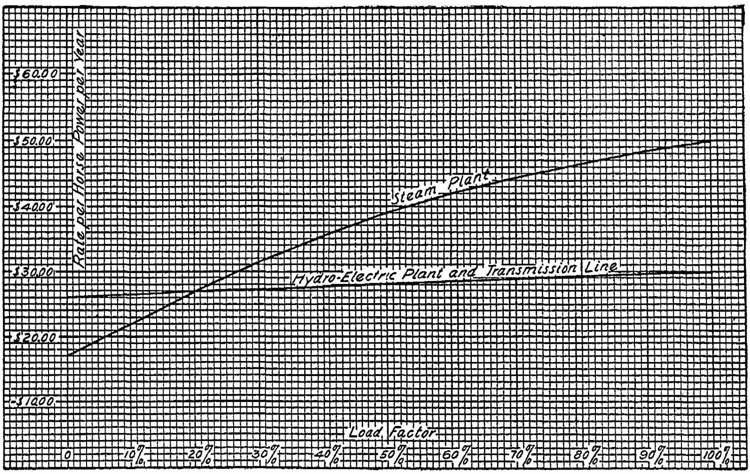


Fig. 219. Method of Plotting Costs per Horsepower per Year in Terms of Load Factor, and Rate per Horsepower per Year

Electric Plant' is intended to represent rate of power cost after transmission for some distance, while that marked 'Steam Plant' represents rate at the power-house switchboard. These curves are about the best obtainable from any power houses of 5000- to 10,000-horsepower capacity, with coal from \$2.50 to \$2.75 per ton."

Fig. 220 shows the same rates as represented in Fig. 219, reduced to cost per horsepower per year, of power actually used; while Fig. 221 gives these same costs in terms of kilowatt-hours, of power actually used.

Since there are approximately 8640 hours per year, and since

1 horsepower is approximately equal to $\frac{3}{4}$ kilowatt, the values in Figs. 220 and 221 are mutually convertible by means of the conversion factor $\frac{3}{4}$ of 8640=6480. Or Fig. 221 may be constructed directly from the values taken from Fig. 219, by means of the conversion diagram, Fig. 222. Fig. 223 is interesting in that it shows in a general way how the fixed charges and operating charges, and therefore the total cost vary with the load factor. While the per cent of the total cost chargeable to operation increases with the load factor, the actual cost of operation per kilowatt-year decreases.

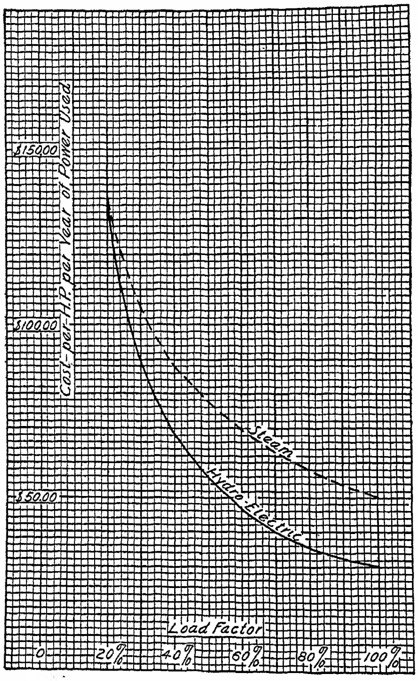


Fig. 220. Rates of Fig. 196 Reduced to Cost per Horsepower per Year, of Power Used

COST LIMITS*

195. General. "Whenever the development of a water power for the purpose of selling water or mechanical or electrical energy is under consideration, the most important question to be decided is: What is the limit of cost per horsepower that may be expended for a development and still leave the plant a financial success; or what is a reasonable price to be charged per horsepower per year?

"A great amount of data has been published in regard to the cost of hydraulic power and power plants; but, as water powers present an infinite variety of conditions, such prices of other plants should be used only with the greatest precaution. A few general figures applying to conditions now prevailing in the northern part of the United States and in Canada may be given here.

196. Unit Costs. "A water-power electric plant, including transmission line and substation, where such are required, but

*Thurso, "Modern Turbine Practice".

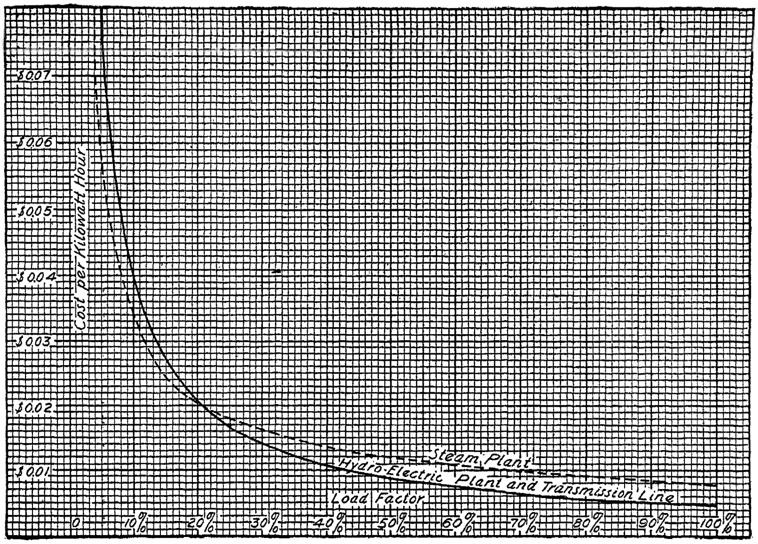


Fig. 221. Rates of Fig. 196 Reduced to Cost per Kilowatt-Hour, of Power Used

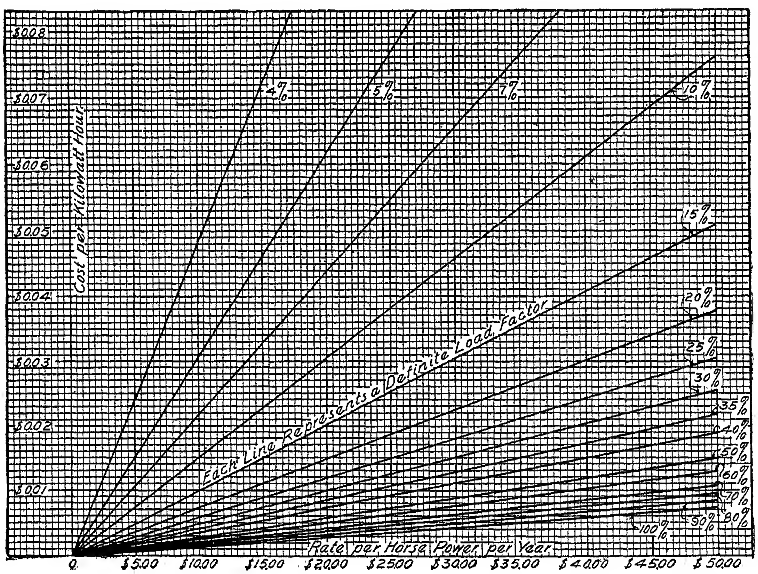


Fig. 222. Method of Changing Rate per Horsepower per Year at Various Load Factors, to Equivalent Cost per Kilowatt-Hour, and *Vice Versa*

without the local distribution, should not cost more than $100 per electrical horsepower, if situated in a remote location or in a farming district; but $150 to $200 may be expended per electrical horsepower for power plant, transmission, and substation, if the power can be sold in a large city or industrial district.

"The price charged for power water per gross horsepower per year, delivered at or near the customer's turbines, may be taken

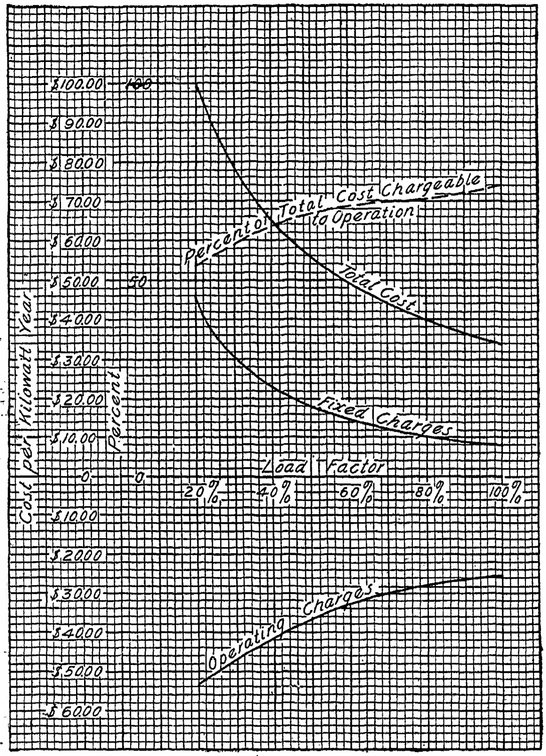


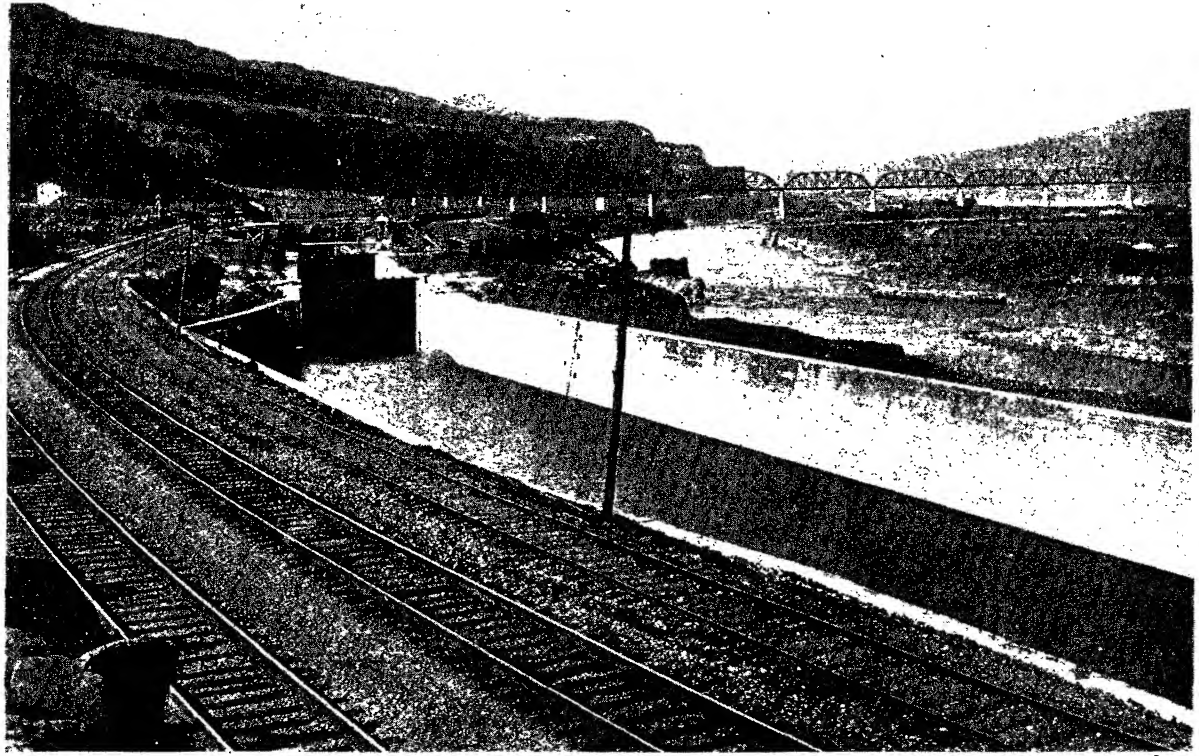
Fig. 223. Curve Sheet of Power Station with 5500-Kilowatt Steam Turbo-Generators

at from $5 to $15, the lower figure being for remote locations, low heads, and large powers, and the higher figure for the reverse conditions. The price of $15 to $25 per mechanical horsepower per year at the power house, or of $25 to $50 per electrical horsepower per year delivered to the customer, may be taken as the limits paid at present. Here, again, the lower rate is for remote locations and large powers, and the higher for the reverse conditions.

"It is also safe to state that in a climate such as that of the northern part of the United States, and in Canada, with the long and severe winters, it does not pay to develop a water power if the power produced will cost more than 75 per cent of the amount for which steam power could be produced in the same locality.

"In Canada, with the great number of water powers yet undeveloped or only partly utilized, it must be regarded as poor policy to install a larger plant than can be run at all stages of the water, or to have a great proportion of power dependent upon storage lakes during the months of low water.

"A water power requiring an auxiliary steam plant during the low-water season, can only pay if either the cost of the development is exceptionally low, or the locality very favorable for the sale of power, or both."



VIEW OF UPPER END OF CELILO CANAL OF THE COLUMBIA RIVER

This canal has eight locks, overcomes a total fall of 82 feet at low water, and costs $6,000,000. The railroad tracks in the foreground are those of the Portage Railroad and the O. W. R. & N. line side by side. The Southern Pacific Railway bridge is shown in the distance.

Photo by Underwood and Underwood, New York City

RIVER AND HARBOR IMPROVEMENT

RIVERS

The term *river improvement* may be said to refer to any engineering work undertaken with a view to bettering the conditions of a stream, either with the thought of facilitating water transportation, or of preventing floods; and, to quote a well-known author on Civil Engineering, "There is no subject falling within the province of the engineer's art, that presents greater difficulties and more uncertain issues." The reason for this will become evident when it is realized upon how many different and complex factors river flow is dependent, and that each of these factors may vary through a very wide range.

Whatever may be the nature of the engineering project undertaken for the improvement of a river, it will have in the main as its object, the removal of obstructions to increase either the depth or width, or both, of the channel; the aiding of navigation, thus permitting the use of vessels of heavier draft; or the protection of the banks to prevent the destructive effect of the freshets which are responsible for these obstructions.

ITEMS OF RIVER SURVEY

Such an undertaking requires a careful and minute study of the characteristics of the river itself; and the more complete this is, the more satisfactorily may the improvements be planned. Where no records exist, the collection of this information is usually made by means of a survey to determine the many and varying factors which may control or help to affect the nature of the improvement. This is usually a difficult undertaking, and may require a knowledge of both nautical and engineering matters. In the former, it is necessary to determine the outline of the channel and its depth, the nature of the banks, currents, tides, winds, etc.; in the latter, there must be ascertained the volume of water discharged at varying

stages, the nature of the bed of the stream, the geology of the watershed, the existence of reservoirs, swamps, river courses, the slope of the land, and in general, what is known as the topography of the country, together with numerous other data that will be elaborated in greater detail farther on. Currents are of greater importance to the engineer than to the navigator, and their determination is the most important and difficult part of an engineering survey. The survey need not necessarily be made with the level, transit, or other engineering instruments, but if not so made, the knowledge must be supplied from maps, charts, statistics, and records, that have been compiled from data so collected.

Specificially, the survey of a river and its watershed would require the investigation of the main stream from its mouth to the source, together with the determination of the flow of entering streams, and should include a complete analysis of the area of the watershed, both as to topography, hydrography, rainfall, run-off, and stream flow. While some of these investigations may more properly come within the province of the water-supply engineer, they nevertheless form the basis for the work that the engineer engaged in river improvement must undertake.

Watershed. The watershed of a river comprises that tributary area from which the rain falling upon it will be directed to the stream. The amount of water flowing in the stream, so far as the watershed is involved, will depend upon several things: (1), and most important, the amount of rainfall precipitated on the watershed or basin; (2) the area of the watershed; and (3) the portion of the rainfall which reaches the stream. The last is controlled or affected by many factors, the more important of which are the evaporation that takes place, the nature and amount of the vegetation covering the watershed, the geology, and the slope of the land.

Topography. The topography of a country or district refers to the relation that exists between hills and valleys, streams, roads, railroads, property lines, fence lines, and such other data, which, when located on a map, will indicate to the observer the changes in elevation of the ground, and the positions of the natural and artificial features of it.

Hydrography. Hydrography concerns itself particularly with the position of streams, their means of supply, volume, currents,

slopes, cross-section, dry season and freshet flow, and other similar characteristics.

Rainfall. The rainfall is pre-eminently the most important of all in its effect upon stream flow. Where this is unknown in a district tributary to a river under investigation, it may be necessary to make an extended series of observations to determine its amount. Usually, however, this will be found to have been done already, since in all civilized countries statistics of the rate of rainfall in different localities are now kept, and have been for a long period of years.

Rainfall follows no set law; and where a river passes through extensive territory, the amount of precipitation is generally as variable as the climatic conditions of the adjacent country; hence, with no data at hand, it might require a determination in each separate district. Such records as these have no value, however, unless kept over a long period of years, as the amount will vary greatly from one year to another, and from one season to the next.

Run-Off. Run-off is a term which refers to that portion of the rainfall which actually finds its way to the streams. No exact method for determining this quantity has been established, as so many variable factors enter into the problem; but it is one of the most important subjects in the analysis of stream flow and Professor D. W. Mead* indicates how numerous and diversified these factors are, in summarizing the matter thus:

The factors that modify or control the quantity and occurrence of the run-off from a drainage area may be briefly summarized as follows:

1. *Precipitation.*
   (a) Whether it occurs as rain or as snow.
   (b) The amount of each, and the total annual precipitation.
   (c) Its distribution throughout the year.
   (d) Its intensity or manner of occurrence.
   (e) The character of storms, including their direction, extent, and duration.
2. *Temperature.*
   (a) The variations of temperature on the area.
   (b) The relation of extreme temperatures to the occurrence of precipitation.
   (c) The accumulation of snow and ice, caused by low temperatures.
   (d) The occurrence of low temperatures causing the freezing of the ground surface at times of heavy spring rains and resulting in excessive run-off.
3. *Topography of Drainage Area.*
   (a) As to whether the surface is level or inclined and the degree of inclination.

* "The Flow of Streams and the Factors that Modify It, with Special Reference to Wisconsin Conditions."

(b) As to the character of area—whether smooth or rugged.

4. *Geology of Drainage Area.*
   (a) Whether pervious or impervious.
   (b) If pervious, whether such pervious deposits are (1) shallow or deep, and (2) level or inclined; whether the outlet or point of discharge of the pervious deposits is (3) in the lower valley of the same river, or (4) in valleys of other rivers, or in the sea.
   (c) As to the condition of the channel of the stream, whether (1) pervious or impervious; (2) whether or not the bed contains more or less extensive deposits of sand and gravel, permitting the development of a more or less extensive underflow.
5. *Condition of Surface.*
   (a) Whether bare or covered with vegetation.
   (b) Whether in natural condition or cultivated.
   (c) Nature of vegetation—whether grassland, cultivated crops, or forests.
6. *Character of Natural Storage on Drainage Area.*
   (a) Nature and extent of surface storage, consisting of lakes, ponds, marshes, or swamps.
   (b) Nature and extent of ground storage, consisting of gravel, sand, or other similar pervious deposits.
7. *Nature of Drainage Area Considered.*
   (a) As to size—whether large or small.
   (b) As to shape—whether long and narrow, or short and broad.
   (c) The location of the area relative to prevailing winds.
   (d) The direction relative to the path of storms.
8. *Character of Stream and Its Tributaries.*
   (a) As to slope or gradient—whether flat or inclined.
   (b) As to falls and rapids on the stream.
   (c) As to the section of the stream—whether deep or shallow.
   (d) As to the arrangement of tributaries—whether joining the main stream at various points along its course, or concentrated in a fanlike arrangement at a more or less common point of discharge.
9. *Artificial Control of Stream.*
   (a) As to dams and storage reservoirs on the drainage area.
   (b) As to the restriction of river sections by dikes and levees.
   (c) As to the obstruction of the stream by piers, abutments, and other encroachments on or adjacent to the waterway.
10. *Artificial Use of Stream.*
    (a) For irrigation.
    (b) For water supply.
    (c) For the supply of navigation canals.
    (d) For artificial storage and its regulation.
11. *Character and Extent of Winds on Drainage Area.*
    (a) As to their intensity and direction.
    (b) As to the modification of them by mountains and forests.
12. *Ice Formation.*
    (a) As modifying the winter flow of the stream.
    (b) As to the formation of ice gorges and their accompanying floods.

FLOW OF RIVERS

With respect to vegetation, it is observed that as the land becomes more thoroughly cleared of timber and underbrush, and the country becomes more thickly populated, the more rapidly does the rainfall reach the river, and the less regular is the flow of the stream. In consequence, rivers passing through such districts approach the condition of torrents at some periods, and are almost dry at others. Forests therefore act as regulators of stream flow, thus helping to prevent floods and to reduce denudation, which latter in turn means greater channel depth in the rivers. In Switzerland, there is such a keen appreciation of the effect on stream flow of growing timber situated on the watershed, that laws have been passed for the promotion and encouragement of its growth on the rugged mountain sides. The particular feature of timber in relation to the regulation of the flow of rivers, is that the leaves and branches overhead produce shade which prevents rapid evaporation or the melting of snow, while the roots and tendrils hold the water and give it off only gradually.

It is not necessary to discuss here in too great detail these or other features relating to stream flow; but it is necessary to point out that such factors have their effect upon the rapidity and amount of river discharge. Besides the necessity of determining such factors as have been mentioned above, it is important to know the nature and amount of the traffic which a river carries, and its relation to the improvements proposed. The velocity of a river and the amount of water it discharges may be estimated approximately by a study such as that indicated in the preceding pages; but there are more usual and satisfactory methods, and they will here receive some consideration.

Before proceeding to this matter, however, we will abstract from the papers of the *U. S. Geological Survey* certain "definitions of terms" and "convenient equivalents" which have a considerable value in a complete understanding of the subject.

Definition of Terms. *Discharge.* The volume of water flowing in a stream—the "run-off" or "discharge"—is expressed in various terms, each of which has become associated with a certain class of work. These terms may be divided into two groups: (1) those which

TABLE I

Conversion of Discharge in Second-Feet per Square Mile into Run-Off in Depth in Inches over the Area

Discharge (sec.-ft. per sq. mi.)	Run-Off (depth in inches)				
	1 Day	28 Days	29 Days	30 Days	31 Days
1	0.03719	1.041	1.079	1.116	1.153
2	.07438	2.083	2.157	2.231	2.306
3	.11157	3.124	3.236	3.347	3.459
4	.14876	4.165	4.314	4.463	4.612
5	.18595	5.207	5.393	5.578	5.764
6	.22314	6.248	6.471	6.694	6.917
7	.26033	7.289	7.550	7.810	8.070
8	.29752	8.331	8.628	8.926	9.223
9	.33471	9.372	9.707	10.041	10.376

Note. For partial month, multiply the values for 1 day by the number of days.

represent a rate of flow, as second-feet, gallons per minute, miner's inches, and discharge in second-feet per square mile; and (2) those which represent the actual quantity of water, as run-off in depth in inches and acre-feet.

Second-Foot. "Second-foot" is an abbreviation for cubic foot per second and is the unit for the rate of discharge of water flowing in a stream 1 foot wide, 1 foot deep, at a rate of 1 foot a second. It is generally used as a fundamental unit from which others are computed by the use of the factors given in the following table of equivalents.

Second-Feet per Square Mile. Second-feet per square mile is the average number of cubic feet of water flowing per second from each square mile of area drained, on the assumption that the run-off is distributed uniformly both as regards time and area.

Run-Off (Depth in Inches). Run-off is the depth to which the drainage area would be covered if all the water flowing from it in a given period were conserved and uniformly distributed on the surface. It is used for comparing run-off with rainfall, the depth of which is usually expressed in inches.

Acre-Foot. An "acre-foot" is equivalent to 43,560 cubic feet, and is the quantity required to cover an acre to the depth of 1 foot. The term is commonly used in connection with storage for irrigation.

Convenient Equivalents. Following are conversion tables and a list of convenient equivalents for use in hydraulic computations.

TABLE II
Conversion of Discharge in Second-Feet into Run-Off in Acre-Feet

Discharge (sec.-ft.)	Run-Off (acre-feet)				
	1 Day	28 Days	29 Days	30 Days	31 Days
1	1.983	55.54	57.52	59.50	61.49
2	3.967	111.1	115.0	119.0	123.0
3	5.950	166.6	172.6	178.5	184.5
4	7.934	222.1	230.1	238.0	246.0
5	9.917	277.7	287.6	297.5	307.4
6	11.90	333.2	345.1	357.0	368.9
7	13.88	388.8	402.6	416.5	430.4
8	15.87	444.3	460.2	476.0	491.9
9	17.85	499.8	517.7	535.5	553.4

NOTE. For partial month, multiply the values for 1 day by the number of days.

METHODS OF DETERMINING FLOW

Actual Measurement. It is customary in the preliminary survey, if the river is small, to select some convenient stretch with a fairly straight course, uniform channel, and regular flow, where a gaging station may be located and observations conducted for the purpose of learning the nature of the profile, the velocity of flow, quantity of discharge, and other like data. This may be accomplished by stretching a wire or rope across the river from bank to bank, at right angles to the axis of the stream, so that soundings with a rod may be taken at intervals along it, and the cross-section thus plotted. The distances between these soundings will depend upon the regularity of the bed; the more uniform it is in outline, the fewer will be the soundings found necessary. A determination of the velocity of the current must also be made at this point, either by means of a weir, by observations on floats, or by the use of the current meter, Fig. 1.

While a scheme of this kind may be perfectly satisfactory for streams of small dimensions, and is, in fact, the method adopted by the *U. S. Geological Survey* in its gaging of streams of this character, it is likely to prove insufficient for a large river. For example, were the river a mile wide and 100 feet deep, it is quite evident that to stretch a rope across would be impossible, as would also the measurement of the depth by a rod. Here, moreover, though they are frequently used, floats prove unreliable, because of contrary

TABLE III
Conversion Equivalents

Second-foot:

1 =	40.	miner's inches (California); law of March 23, 1901
1 =	38.4	miner's inches (Colorado)
1 =	40.	miner's inches (Arizona)
1 =	7.48	United States gallons per second
1 =	448.8	United States gallons per minute
1 =	646,317.	United States gallons per day
1 =	1.	acre-inch per hour (about)
1 (for 1 year)		= 1 square mile, 1.131 feet (13.572 inches) in depth
1 (for 1 year)		= 31,536,000 cubic feet
1 (for 1 day)		= 86,400 cubic feet

Cubic feet:

1,000,000,000 (1 United States billion)	= 11,570 second-feet, for a day
1,000,000,000	= 414 second-feet, for 1 28-day month
1,000,000,000	= 399 second-feet, for 1 29-day month
1,000,000,000	= 386 second-feet, for 1 30-day month
1,000,000,000	= 373 second-feet, for 1 31-day month
1,000,000	= 22.95 acre-feet
1	= 0.0283 cubic meter
(of water) 1	= 62.5 pounds

Miner's inches:	
100 (California)	= 18.7 United States gallons per second
100 (California), for 1 day	= 4.96 acre-feet
100 (Colorado)	= 2.60 second-feet
100 (Colorado), for 1 day	= 5.17 acre-feet
100 (Colorado)	= 19.5 United States gallons per second
Gallons (United States):	
100 per minute	= 0.223 second-foot
100 per minute (for 1 day)	= 0,442 acre-foot
1,000,000	= 3.07 acre-feet
1,000,000 per day	= 1.55 second-feet
Acre-foot:	
1	= 325,850 gallons
Inch deep, on 1 square mile:	
1	= 2,323,200 cubic feet
1	= 0.0737 second-foot per year
Foot:	
1	= 0.3048 meter
Mile:	
1	= 1.60935 kilometers
1	= 5280 feet
Acre:	
1	= 0.4047 hectare
1	= 43,560 square feet
1	= 209 feet square (nearly)
Square mile:	
1	= 2.59 square kilometers
Cubic meter per minute:	
1	= 0.5886 second-foot
Horsepower:	
1	= 550 foot-pounds per second
1	= 76 kilogrammeters per second
1	= 746 watts
1	= 1 second-foot falling 8.80 feet
$1\frac{1}{3}$	= 1 kilowatt (about)

To calculate water power quickly:

(Second-feet × fall in feet) ÷ 11 = net horsepower

(on a water wheel realizing 80 per cent of the theoretical power)

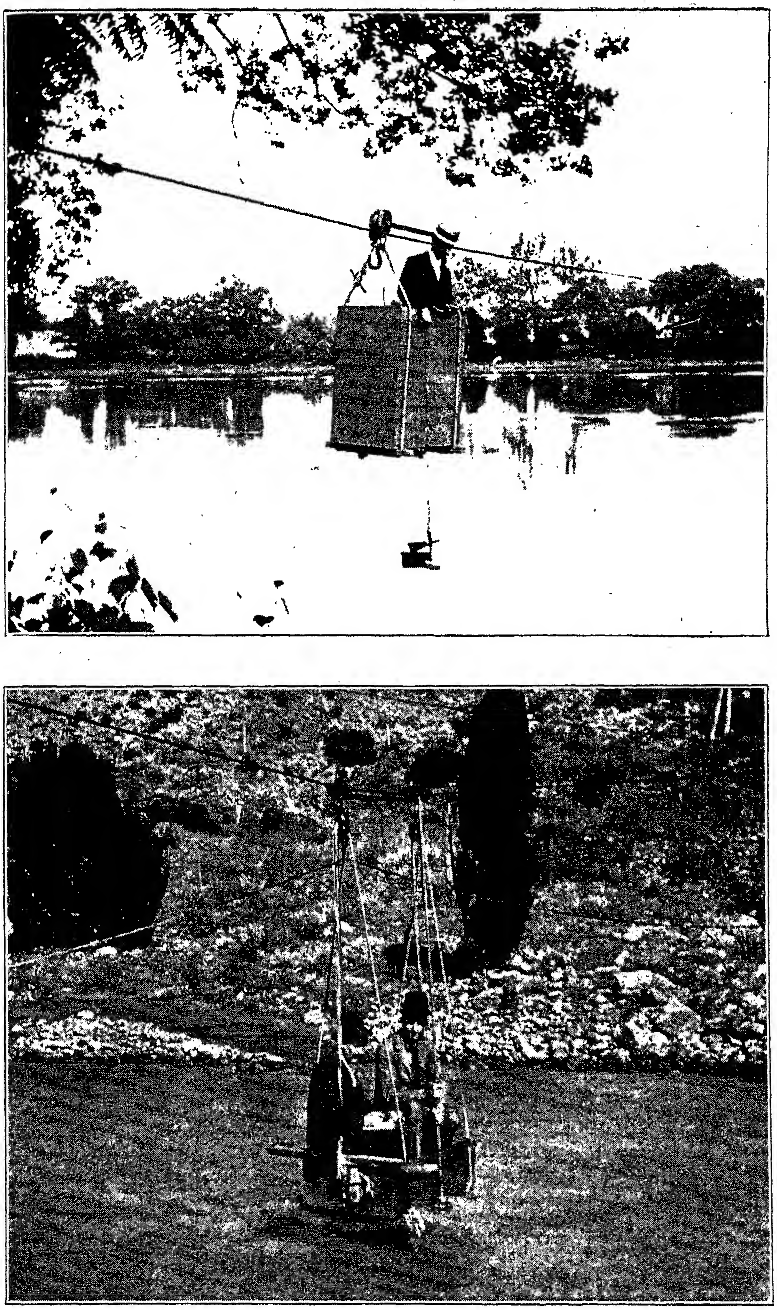


Fig. 1. Types of Cars Used in Measuring Velocity of Streams
Courtesy of U. S. Geological Survey

currents; and the measurement of the fall is difficult to obtain, because of surface undulations or local elevations arising from the curvature of the bed or the effect of the wind. Other methods, therefore, must be employed to determine the profile, while the gaging of the velocity should be undertaken by means of the current meter.

To determine accurately the cross-section of a large river, it is usual to establish transit stations on the shores in known positions, and from these to take observations on a boat carrying the sounding party, by measuring the horizontal angle from the boat to some base line on shore. The sounding party consists of several members—one to row, one to heave the lead and read the depth on the chain, and a third to keep notes and to signal to the transitman on shore so that he may sight on the flag and thus locate the spot in the river where the sounding is being taken. Various modifications of this general method occur, but the principles are the same.

Use of Formulas. *Chezy's Formula.* Under some circumstances it may be deemed advisable to dispense with the above methods for the determination of velocity, and resort is then had to the use of formulas. In this respect, the one best known is that deduced by Chezy in 1775, after a thorough analysis of all available data. The formula

$$v = c\sqrt{rs}$$

is one for the mean or average velocity, and the quantities appearing in it have the following significance: c equals an experimental coefficient depending upon the nature of the river bed; r equals hydraulic radius; and s equals slope equals $\frac{h}{l}$ equals height divided by length.

"The hydraulic radius of a water section is its area divided by the wetted perimeter, and the wetted perimeter of the cross-section of a channel is that part of its boundary which is in contact with the water."

Kutter's Formula. It is quite evident that the important quantity in the determination of the mean velocity by this formula is the factor c, since r and s are easily measured; and hence the accuracy of c will determine the reliability of the formula in expressing the approximate mean velocity.

The factor c is generally derived from what is known as Kutter's formula, in which

$$c=\frac{\frac{1.811}{n}+41.65+\frac{0.00281}{s}}{1+\frac{n}{\sqrt{r}}\left(41.65+\frac{0.00281}{s}\right)}$$

and in which it is seen that c is made to depend upon r and s, and also upon a quantity n which is called the *friction factor,* and which expresses numerically the condition of the roughness of the channel.

Gaging by Floats. In most cases, however, it is more expedient and much more satisfactory, particularly in the case of large streams, to gage the flow by means of floats of various types, or by the use of a current meter. Floats were extensively employed on the Mississippi

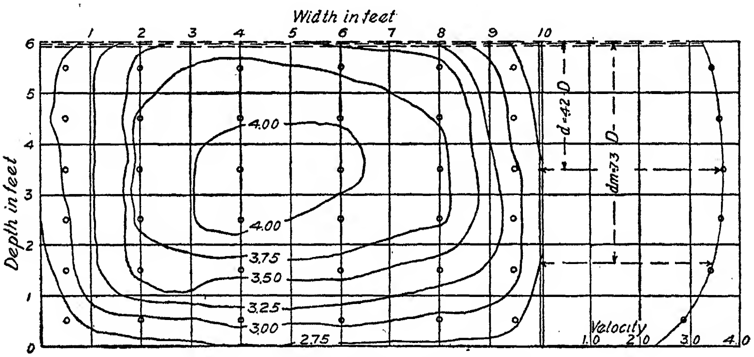


Fig. 2. Velocity Curves in Cross-Section

River in some of the earlier surveys, where elaborate pains were taken to secure accurate values.

Where either floats or the meter are used, the vertical cross-section of the river at right angles to its axis is divided into vertical strips of such width that the velocity in each may be in approximate conformity with the general conditions of flow of the stream. The various velocities in each one of these strips is measured; or at least such velocities are measured as may lead to the deduction of a mean velocity; and the sum of these mean velocities for all the strips, divided by the number of sections, will give the mean velocity for the entire cross-section, Fig. 2.

It should be stated that usually, in the measurement of streams, it is the mean velocity that is desired, or the velocity which would

result if the discharge were to be divided by the area of the cross-section. The mean velocity is, however, a difficult quantity to determine; for not only do the velocities vary on the surface in going from shore to shore, but they also vary between the surface and the bottom.

The greatest surface velocity will usually be found over that part of the channel where it is deepest; while the greatest velocity in any vertical section between the surface and the bottom, will be found at a point below the surface about $\frac{4}{10}$ of the total depth at that point. The variation in the surface velocity is due to the greater friction that the water encounters near the banks than in mid-stream; while the variation in velocity in a vertical section is due to the greater friction met with by the lower layers of water in flowing over the bed, as compared with that of those layers above, which flow merely over other layers of water. For this reason, and because they feel the effects of winds, currents, eddies, etc., surface floats will give only approximate values, even if the observations be taken in various sections of the river.

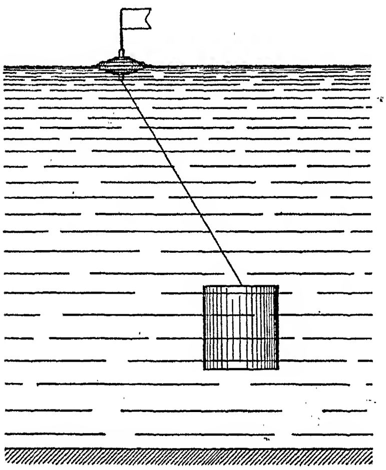


Fig. 3. Mid-Depth Float

Mid-Depth Float. Mid-depth floats are very much better, since they are unaffected by the wind, and record a more normal velocity. Such a float consists of two parts: that which is submerged and is carried by the current; and that attached to the former by a thin cord or wire, and floating on the surface, thus serving as an indicator to locate the part submerged. The wire or cord should be as thin as possible, to avoid any loss of velocity due to friction; and the surface attachment should be as small as may be permissible under the circumstances, Fig. 3.

Neither the surface nor the mid-depth float gives the mean velocity of flow; but formulas may be applied which will reduce these observations to the quantity desired, though the result will be but an approximate value.

Rod Float. Another form of float is that known as the rod float. It consists of a hollow rod or cylinder of tin, and is weighted with stone or shot, so as to stand vertical in the water, immersed to a depth almost equal to that of the stream. As this float almost reaches the bottom, its speed will represent the average velocity of all those

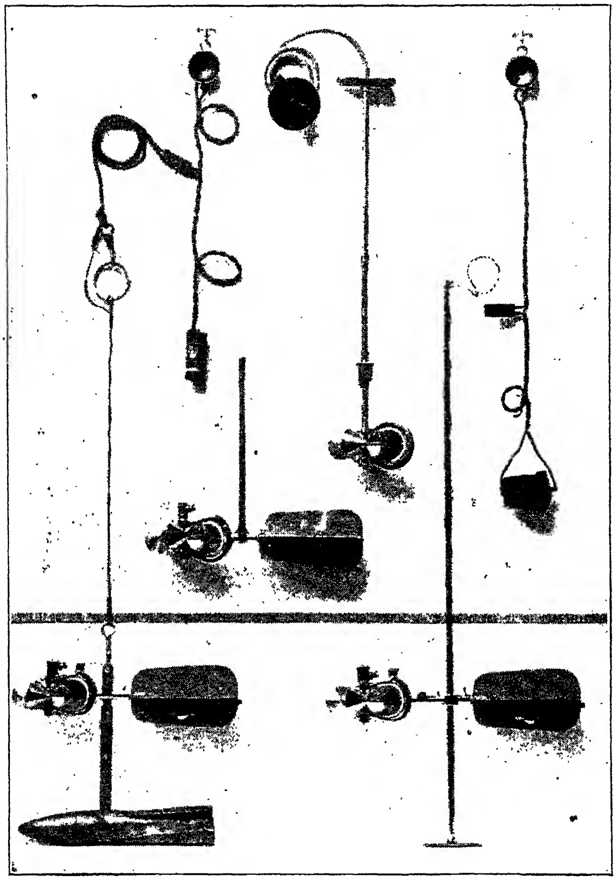


Fig. 4. Types of Price Current Meters
Courtesy of U. S. Geological Survey

taken in a vertical plane. An observation on any of these types of floats is made simply by noting the time required for it to pass a given length of the stream.

Current Meter. Current meters are better adapted to the conditions found in large rivers, and they give more accurate results than

do floats. The meter is constructed in various forms, but generally consists of a set of revolving cups attached to a vertical rod, Fig. 4. These are made to point upstream by a vane or tail, Fig. 5, which trails behind them, so that as the water strikes the cups they will be

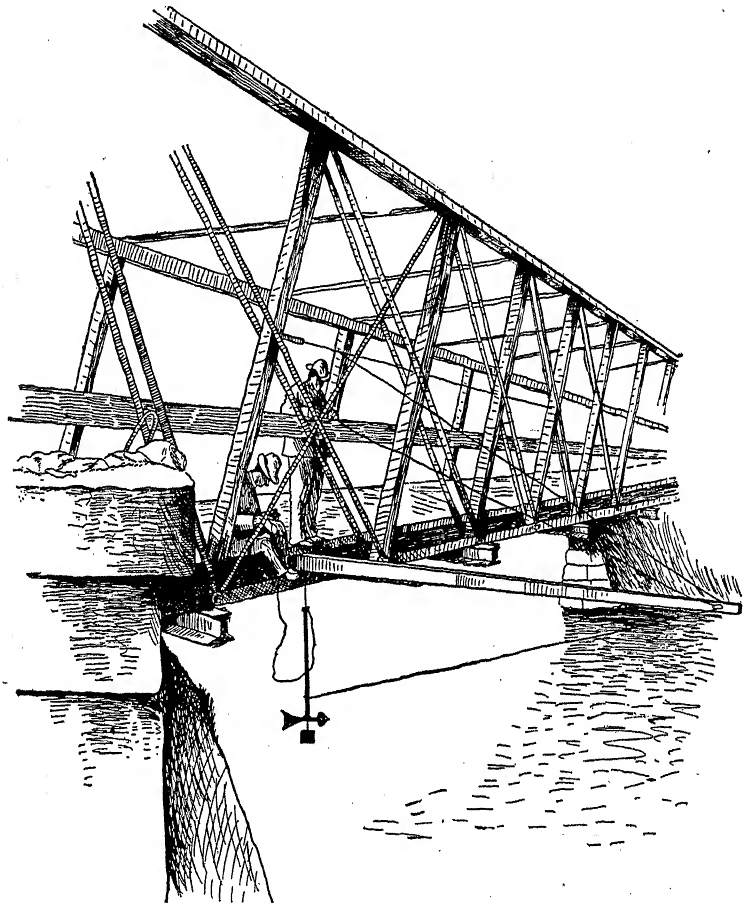


Fig. 5. Device for Holding Current Meter in Position

made to revolve. This revolution is recorded by an automatic electrical instrument, Fig. 6, having recording drums or pointers, which indicate the speed. Readings are taken systematically throughout the entire cross-section, and from this record a mean velocity is calculated. This question of velocity is of prime importance in the

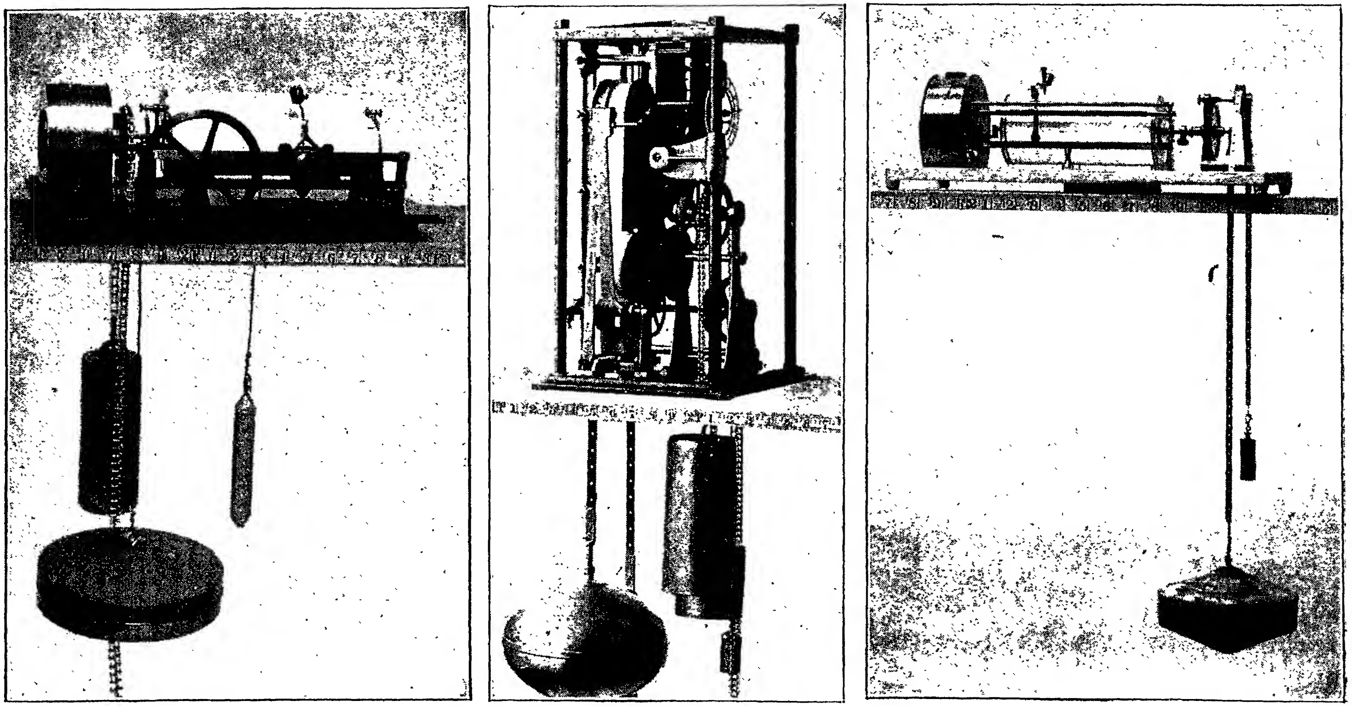


Fig. 6. Types of Automatic Gages—Left, Stevens; Center, Gurley; Right, Friez
Courtesy of U. S. Geological Survey

discussion of river improvements, since upon it depend the transporting and eroding powers of the river.

Weir. It should be stated that the preceding methods of stream measurement apply to those of large volume. Where the discharge is comparatively small, or where conditions are favorable, Fig. 7, a weir may be used, however. "A weir is a notch in the top of the vertical side of a vessel or reservoir through which water flows. The notch is generally rectangular and the lower edge truly horizontal, with sides vertical. The lower edge of the rectangle is called the crest of the weir."

When a weir is employed to measure a stream, empirical formulas are used to compute the velocity, so that it is perhaps the most accurate method applicable to small streams. The student is referred for more detailed information to the textbook on "Hydraulics" or to any of the many other standard textbooks on the subject.

FLOODS

Affecting Conditions. The foregoing remarks on the flow of rivers refer primarily to the normal stage; but it should be remembered that any improvement which is undertaken, and is desired to remain as a permanent piece of work, must be designed to withstand the extreme condition of flood, as well as the less severe one of normal flow.

All streams vary in the amount of water they discharge, both from year to year and from season to season. Some have less of a variation than others, as in the cases of those fed by springs or from lakes where the volume may be practically constant. In the cases of other streams, however, at one time they may be torrential, at another dry river courses. Because of this variation, a stream is alternately large and small, and, in consequence, at the varying stages it has varying tendencies. The features of the channel are, however, controlled by the flood discharge.

Rainfall. Floods depend principally upon the amount, suddenness, and duration of the rainfall; but the nature of the watershed with respect to geology, vegetation, slope, etc., also has an important bearing on the subject, so that all those factors which have been referred to as affecting the flow of rivers may be regarded with equal interest in the discussion of the subject of floods.

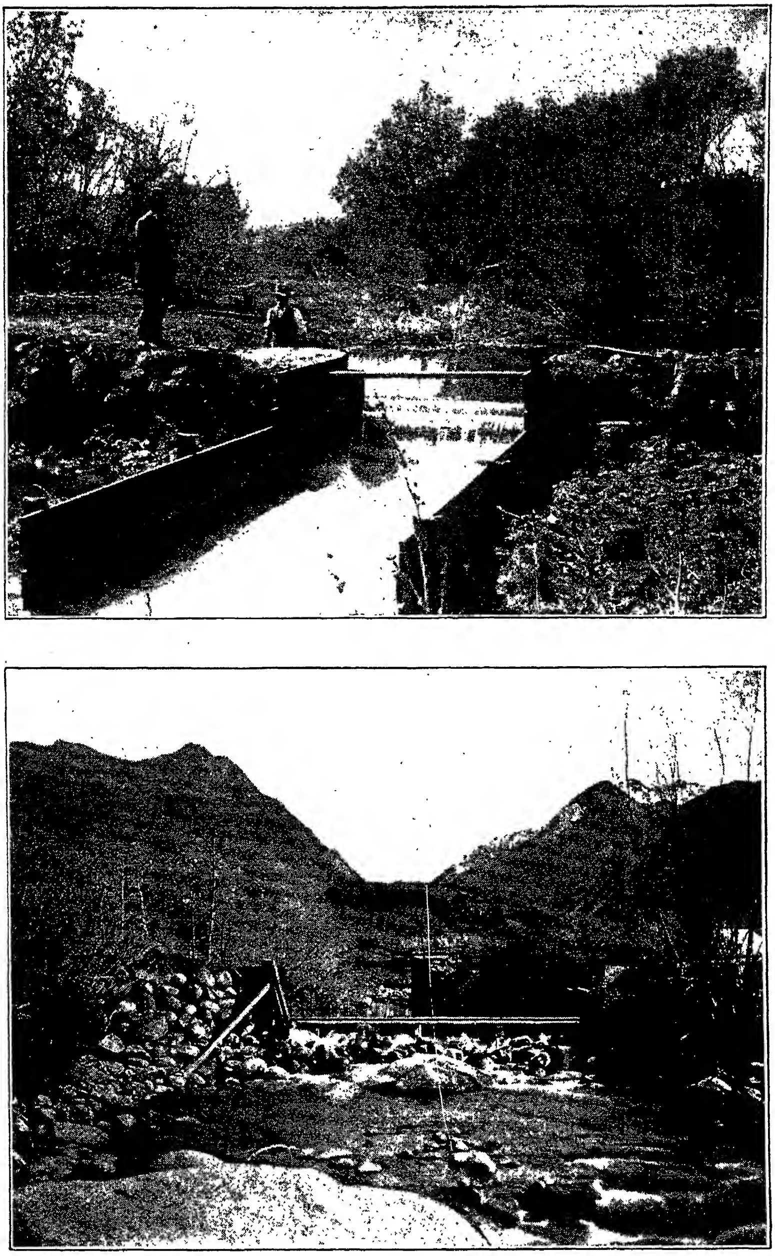


Fig. 7. Types of Measuring Weirs—Top, Los Angeles River, California; Bottom, Cippoletti Weir on Little Cottonwood Creek, Utah
Courtesy of U. S. Geological Survey

Watershed. The size of the watershed is peculiarly important, as the larger the area, generally speaking, the less will be the discharge in the river, per unit of area. Thus it has been observed that, while on small watersheds the discharge may be 260 cubic feet per second per square mile, on extensive watersheds of 5000 square miles or more it may be as small as 30 cubic feet per second per square mile.

Seasons. When an improvement of a river is contemplated, it is necessary to know at what seasons of the year the floods are likely to occur, the amount of water they carry, how long they take to

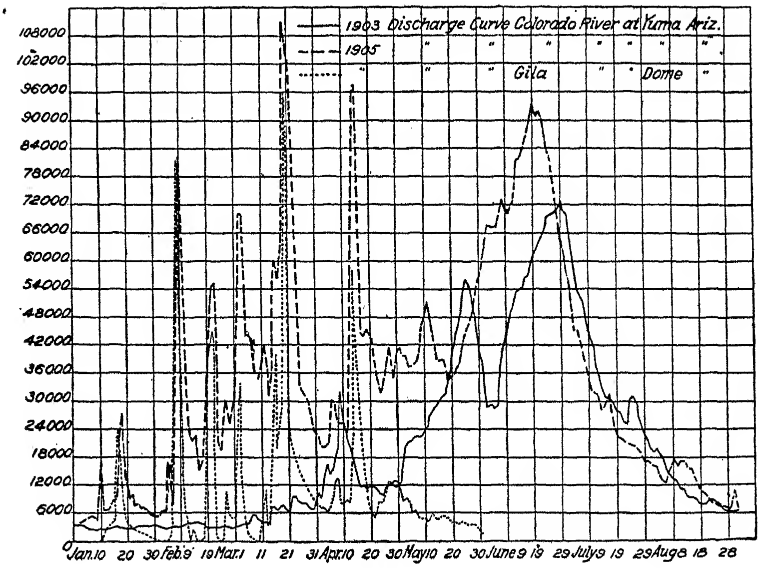


Fig. 8. Diagram Showing Flood of Colorado and Gila Rivers

develop, how long they last, and any other characteristics, so that precautionary methods may be adopted in the design of the protective structure to insure its withstanding the destructive effects of the water. At the same time, similar data should be collected on the dry-weather or minimum flow and the frequency with which these conditions occur, to study in turn its possible effect upon the protective works.

In tropical climates, where dry and wet seasons alternate, these variations in the flow of a river are not only much more common but also much more pronounced than in a cold or temperate region, and the streams are quite likely to vary in their flow from a small

one to a raging torrent. A notable example of this type of river is found in the Colorado River of our own country, one of its chief characteristics being extreme variability of flow, Fig. 8.

The most serious floods are naturally those of the spring season, and are generally brought about by a heavy rain falling on melting snow. A similar condition may result when a heavy rainfall occurs during a very short period of time.

Channel. Under normal conditions, the channel of a river is eminently fitted to care for the amount of water it is required to discharge; but at flood periods, unless the banks are high or well protected, the river is likely to overflow and to cause destruction of life and property in the vicinity. A large part of the work of river improvement concerns itself, therefore, with the study of means to protect the banks so that this loss may be avoided.

In many cases, flood conditions will have but little effect upon navigation—rather aiding it, in fact, than hindering it, by furnishing a greater depth and width of channel. On the other hand, however, currents are sometimes produced by floods which make navigation dangerous; besides which, the elevation of the water may be so great that the passage of boats beneath bridges may be made impossible.

Floods, unless confined to the river's channel, may cause great damage; but where so confined and controlled, they may serve, on the other hand, to remove from the bed of the stream the sediment which has been deposited there by the slower-moving normal current. These destructive floods frequently occur along the Ohio and Mississippi rivers. Thus, the Ohio has experienced several floods of 50 feet in height; and in 1884, the river at Cincinnati rose to an elevation of over 70 feet.

FLOOD CONTROL

Means of Protection. To prevent flood damage, provision must be made to care for the surplus water in some satisfactory manner. On the lower Mississippi where these destructive risings have done damage probably in excess of that of any other river in the world, and where, in consequence, the subject has received careful consideration, the three following methods have been suggested as a means of protecting the bottom lands and thus avoiding the losses consequent upon overflow: (1) To modify the actual relations existing between the accelerating and retarding forces in the channel in such a manner

as to enable the former to carry off the surplus water without so great a rise in the surface—to this class of protective works belong *cut-offs;* (2) to reduce the maximum discharge by diverting the excess of water to *tributaries, artificial reservoirs,* and *artificial outlets;* and (3) to confine the water to the channel and allow it to regulate its own discharge—such a procedure as this requires *embankments* or *levees.*

The methods generally adopted are levees, and artificial reservoirs. Accordingly they will receive some consideration.

Cut-Offs. A *cut-off* or *straight cut* is either a natural or artificially constructed by-pass, which diverts the water in the main channel from the upper side to the lower side of a bend. It is evident that such an artifice reduces the length of channel, lowers the surface of the water below the point where it first enters the cut-off, and thus increasing the fall, helps to remove the sediment deposited on the bend in the main stream. Cut-offs have been in operation on the Mississippi and its tributaries, either by the action of the river itself or by design; thus, "during the flood of 1903 the distance around the bend at Leland's Neck, near Greenville on the lower Mississippi, was 13 miles, while across the neck was only 3000 feet. The difference in the elevation of water surfaces at the upper and lower sides was 4.7 feet, giving a rate of fall, when the water overtopped the bank, of some 9 feet per mile."

With such means of control, it is necessary to make the cut-offs continuous from the mouth of the river to the point where it is proposed to give relief, for, where the water from the cut-off re-enters the main channel, an increased flow again results, and farther downstream it will be as necessary to introduce cuts as it was above, unless the river itself is capable of handling this flood flow.

Diversion to Tributaries. Diverting the flow of floods to tributaries has often been suggested as a means of getting rid of surplus water, but it has never been applied on a large scale. There are several locations along the Mississippi where the application is quite feasible, but other means have been employed in view of the great expense attached to the undertaking. The plan would generally require the construction of a long diverting canal or channel, and this, owing to the fact that different drainage areas are generally at some distance apart or else separated by intervening hills, would ordinarily increase the cost of construction to a prohibitive amount.

With regard to the upper Mississippi, however, it has been proposed that such a channel be cut through the prairie between the "Great Bend" and the Mouse River, which drains into another system—that of the Red River of the North. This would direct the flood flow of the Missouri to the northward rather than allow it to enter the Mississippi.

There are several factors to be taken into consideration before such a scheme as this can be pronounced a success: (1) the item of expense that would attach to the construction of such a canal, which in the above-mentioned case would be 40 miles long; (2) the difference in elevation between the two streams, which of necessity would have to be sufficient to guarantee a flow in the desired direction; and (3) the periods at which maximum flood occurs in each river—to see that such times do not coincide, giving the greatest flow in each at the same time.

Natural Storage Reservoirs. The application of storage on the headwaters of a stream against flood periods furnishes a legitimate means of taking care of the surplus water of the flood period, as by this expedient it may be impounded when least needed, and gradually fed to the watercourses when their discharge becomes diminished.

Preferably since such reservoirs can usually be constructed only at a very great expense, they should be located in the naturally formed pockets of the hills, along the upper levels of the river, and near the source; for here Nature makes the construction of a dam to hold back the flood flow a comparatively easy matter. On the other hand, near the mouth, the land adjacent to a river is generally flat, so that fewer opportunities for a reservoir site present themselves. With such reservoirs, the water on the lower part of the basin has an opportunity to discharge into the main stream without the addition of the excess from above, and hence the river is better able to handle it.

It must be appreciated that Nature provides storage reservoirs of various kinds such as lakes, ponds, swamps, marshes, or even permeable soils or rocks, which will absorb or hold large quantities of water and feed it out slowly, thus regulating the flow of the rivers and retarding the flood flow.

The following abstract, because of the completeness of the treatment, is given in some detail. It is taken from the *U. S. Government*

Report on the subject of "Storage Reservoirs", made in 1898 by Captain H. S. Chittenden:

Nature presents abundant examples of the effective control of stream flow through the agency of reservoirs. There are, indeed, comparatively few streams whose flow is wholly uninfluenced by such action. The most perfect example in the world, both as to magnitude of the stream and the completeness of control, is the St. Lawrence River, embracing the great chain of North American Lakes. Considering only that portion of the system which lies above the Falls of Niagara, let the flow at the outlet be compared with that of other streams of similar magnitude. For this purpose, take the Niagara River at Buffalo; the Ohio at Paducah, Kentucky; the Missouri at its mouth; and the Mississippi just above the mouth of the Missouri. The following tabulation gives the area of watershed in square miles, and the mean annual discharge in cubic feet per second, of each:

	Niagara	Ohio	Missouri	Mississippi
Watershed (sq. mi.)	265,095	205,750	530,810	171,570
Discharge (cu. ft. per sec.)	232,800	307,000	100,000	130,000

The maximum and minimum discharges, except for the Niagara, show as much greater divergence, the ratios of maximum discharge to minimum discharge for 1883 being as follows: Niagara 1.19; Ohio 28.22; Missouri 29; and upper Mississippi 10.29.

This striking dissimilarity in the regimen of streams of similar magnitude, and, with one exception, of similar climatic conditions, is entirely due to the reservoir action of the Great Lakes.

The mean annual fluctuation of Lake Superior, Lake Michigan, Lake Huron, and Lake Erie, represents an annual storage of 2419 billion cubic feet of water, equivalent to above 153,000 cubic feet per second for a period of 6 months.

In addition to the annual fluctuation, there is constantly going on a periodic change which often requires several years to complete the cycle. As an illustration of this characteristic, take the period of 8 years from 1872 to 1879, during which the mean annual level of the four upper lakes rose for a period of 4 years and fell during the following 3 years. The total storage represented by this rise of mean level was 4000 billion cubic feet. The fall in mean level following the rise was equivalent to 3627 billion cubic feet. After this fall, the mean level began to rise again.

The foregoing figures convey some faint idea of the magnitude of the storage of the Great Lakes, and of the way in which it operates to preserve a balance, not only between the wet and dry seasons of each year, but between those cycles of wet and dry years which are continually recurring. These reservoirs absorb the flood waters of spring, and pay them out in the following dry season, thus preventing floods on the one hand and low water on the other. And while these seasonal changes are going on, the lakes respond to the varying conditions of longer periods, levying upon years of more than average precipitation in order to maintain a flow in the outlets during the years of deficiency which are certain to follow.

The result of this storage action of the Great Lakes is to produce a river system radically different in its general characteristics from nearly all other streams. Such conditions as high and low water, as elsewhere understood,

are here entirely unknown. Commerce pursues its way through these lakes and rivers without serious hindrance except when ice closes the way; and the river and harbor engineer has little to do with low-water problems or protection against floods, but rather with the deepening of harbors and connecting channels for an ever-increasing size of vessels and volume of commerce.

Artificial Storage Reservoirs. While it is impracticable to imitate Nature on the scale of her own work in the construction of reservoirs, her example has nevertheless been followed very extensively on a smaller scale. In fact, works of this character have been built for a variety of purposes since the remotest antiquity. Many reservoirs have had as a prominent reason for their construction the prevention of floods in the valleys below them, although this has seldom if ever been an exclusive reason. In all examples of reservoir construction, however, the purpose has been to correct the inequalities of Nature—to prevent the rapid and destructive flow of rivers at seasons when not needed, and to augment and reinforce that flow when the need does exist.

Mississippi Headwaters. The largest artificial-reservoir system ever yet constructed is that at the headwaters of the Mississippi River. It is dotted with an immense number of lakes—the total number having been estimated as high as a thousand—and 5 reservoirs having a total capacity of 93 billion cubic feet of water, used chiefly to add depth to the river during the period of navigation. Some of the larger of these afford exceptionally favorable opportunities for the inexpensive storage of water. The dams required are low structures; but the area over which the water is raised by them is so extensive that the cost per unit of volume stored is probably the smallest ever yet realized. These remarkably favorable natural conditions for the storage of water have long attracted public attention; and in 1881 actual construction was begun.

The effect upon the navigable stage of the river would, of course, vary with the locality considered, and would diminish rapidly with the distance downstream. But considering that such an improvement is of the most permanent character, depending upon the maintenance of the dams for its perpetuity, the above cost cannot be considered excessive when compared with the vast outlay for the more temporary improvement of these rivers by present methods. A permanent increment of from 10,000 to 20,000 cubic feet per second to the low-water stage of even so large a stream as the Mississippi River is not to be passed over as a matter of small importance.

Flood Protection. Every reservoir built along the course of a stream is, to some degree, a protection against floods in the valley below. The extent of this protection depends, of course, almost entirely on the ratio of its capacity to the flood discharge. A reservoir that can store the entire flow of a stream is an absolute protection against floods for a considerable distance below. It is difficult to propose any general rule for the extent of this control; but, assuming a general similarity of watershed, it would seem not unreasonable to say that it ought to be decisive to at least such a distance below as will give an additional watershed to a stream equal to twice that above the reservoir. In other words, the reduction of a flood wave by $\frac{1}{3}$ of its volume will, in general, rob it of its destructive character.

But in a great many cases this control extends very much farther. For example, in the case of a flood caused by the rapid melting of snows in the mountains, reservoirs situated below, capable of impounding this flood, will protect the entire valley as far as its destructive influence would otherwise have reached. When it is remembered that the volume of a destructive flood is only a part—probably always less than half of the total flow of a year—it will be admitted that a storage capacity equal to one-fourth of the run-off, well distributed throughout a watershed, will practically eliminate the evil effects of floods on its streams, and supply a percentage sufficient for the purposes of irrigation.

It is not necessary, though important, that a reservoir should be empty when a flood comes. Even if full, it still moderates the flow of the stream below, the effect varying directly with the superficial area of the reservoir when full, and inversely with the capacity of the spillway. In this respect it acts precisely as does a natural lake. For example, if the spillway of a reservoir or the outlet of a natural lake be of such an increase of discharge, every increment of this depth of outlet means also an increment of the same depth over the entire reservoir. A flood passing such a reservoir will be reduced by the storage resulting from this increment; and before it can produce a full discharge, it must fill the reservoir to the necessary height above the bottom of the spillway. Therefore, a large reservoir, even when full, is always a perfect protection against sudden floods. In the case of long-continued floods, it greatly retards the arrival of maximum effect, and gives ample notice of its approach.

In fact, this is a very important feature of reservoir action, even where the capacity of the reservoir is not sufficient to prevent the flood entirely. It does prevent freshets—that is, sudden floods—and in smaller streams it is often the suddenness quite as much as the magnitude of floods that causes damage and loss of life. A reservoir ceases to be any protection if a flood continues long enough to fill it to such a height that the discharge at the outlet is equal to the entire inflow.

Combination Discharge. In the case of floods, which are the results of combinations of discharges from the various tributaries, reservoirs may actually operate to increase the combination. Take for example, the natural reservoirs at the sources of the Mississippi. While they restrain the flood excess in that stream, they keep up a heavy flow for some time after the flood has passed. If this larger flow happens to come in with a flood crest at the junction of some tributary below, it will actually increase the combination over what would have been the case without the reservoirs. In the French investigations, the dams proposed for restraining floods were to have open sluiceways without means of closing them. In the ordinary flow of the stream all the water could pass through; but the sluiceways were to be so proportioned that when the flow should pass a certain point, the surplus would be retained in the reservoir, the outflow being always limited by the capacity of the open sluices. The arrangement was, therefore, precisely like that of a natural lake without a dam across the outlet. The outflow could never be entirely restrained, and it would increase in proportion to the height of water in the reservoir.

Now, in the case of a large stream, where flood combination is the really dangerous thing, it was found that these reservoirs, had they actually been constructed, would have increased certain floods. They would have maintained a heavy retard flow on some tributaries, which in their natural condition would have entirely run out before the arrival of floods from other tributaries. As it happened, this retard flow in the one case would have come upon a flood crest in the other, and would actually have increased the natural combination. This, of course, could not be true of reservoirs with closed sluices, unless, as above stated, the reservoirs were entirely filled with the flood passing over them.

It is therefore clear that the efficiency of reservoirs in moderating great floods would have to be a matter of judicious management in controlling combinations, quite as much as of actual capacity.

Motives in Construction. For weighty reasons, very few, if any, reservoirs have been built for the exclusive purpose of protection against floods in the valleys below them; but there are numerous examples where this has been an important consideration in their construction. Various works have been constructed in Europe, but all have other motives, in addition to that of flood protection, to justify their construction. The systematic creation of a comprehensive system of reservoirs on any river for the sole purpose of mitigating the severity of floods, has never been undertaken. The subject has, however, received exhaustive study; and examples of such studies are to be found in France.

"Particular emphasis should be placed upon these studies, because they disclose the true obstacle to the use of reservoirs for the sole purpose of flood prevention. It is the cost, not the physical difficulties, which stands in the way. It may be stated that, as a general rule, a sufficient amount of storage can be artificially created in the valley of any stream to rob its floods of their destructive character; but it is equally true that the benefits to be gained will not ordinarily justify the cost.

"The reason for this is plain. Floods are only occasional calamities, at worst. Probably, in the case of the majority of streams, destructive floods do not occur, on the average, oftener than once in 5 years. Every reservoir built for the purpose of flood protection alone would mean the dedication of so much land to a condition of permanent overflow, in order that three or four times as much might be redeemed from occasional overflow. One acre permanently inundated to rescue 3 or 4 acres from inundation of a few weeks once in 3 or 4 years, and this at a great cost, could not be considered a wise proceeding, no matter how practicable it might be from engineering considerations alone. The cost, coupled with the loss of so much land to industrial uses, would be far greater than that of levees or other methods of flood protection.

"In fact, examples of natural reservoirs, while they show conclusively the vast beneficial influences of large reservoirs upon the flow of streams, also disclose the fatal obstacle to their successful

imitation by man. In only very few places has Nature prepared sites where man can erect works which will create large bodies of water; and even if she had done so, the gain from utilizing them would not equal the loss. The reservoir system of the Great Lakes involves the perpetual withdrawal from agricultural and industrial uses of an area nearly twice the size of the State of New York. It will be found in general, that the surface of the earth where reservoirs could be built on an extensive scale is liable to be of more value in its present condition than it ever could be if covered with water.

"The construction of reservoirs for flood protection is not, therefore, to be expected, except where the reservoir is to serve some other purpose as well; and inasmuch as such purposes are not ordinarily extensive enough to develop systems of reservoirs, upon which, rather than upon isolated works, the control of great floods depends, this large control is hardly one of the possibilities of the future. The only probable exception is that of a reservoir system on the watershed of the Missouri River.

"For flood protection in isolated cases, however, on a relatively small scale, reservoirs will undoubtedly continue to be built, particularly when they serve other purposes as well. From this point of view, they will always be projects of public importance."

Artificial Outlets. This method consists in reducing the flow of water in the main stream by conveying it to the mouth through other channels. It is claimed by some engineers, that such a reduction of flow in the main stream would increase the amount of sediment deposited in the channel, and thus, instead of depressing it, only tend to raise it and the surface level of the river. On the other hand, investigations made on the Mississippi River Survey tend to contradict this, and show that no such deposition of material occurs.

On that river, such outlets have been practically established by the water breaking through the banks or levees (which breaks are called *crevasses*) and being allowed to drain off into the adjoining, low-lying, swamp land. Here it is not so much a question of how to get rid of the water from the river, but what to do with it when it has reached the neighboring bayous and swamps, for the natural drainage proves insufficient, and the backwater rises until plantations are threatened. It therefore requires a special channel to tidewater; and this usually means a prohibition, due to excessive cost. In this

particular instance, therefore, it was deemed very much better to protect adjoining lands from floods by an extensive system of levees; and to this end such a system has been and is being constructed.

Levees. Levees are banks of earth built on or near the shore line of rivers or harbors to prevent the inroad of water. This method of protection was used by the Romans, Egyptians, and other nations of antiquity; while today, wherever rivers are found, the same principles are applied to keep the waters from overflowing the banks or shore. Those countries most actively engaged in this sort of engineering work are the United States, England, France, Spain, Italy, Germany, and Holland. In the latter country, these protective works are called *dikes*, but the significance of this term must not be confused with the later application of the word.

In the United States, the most important application of this principle of engineering is found in connection with the Mississippi River. Until 1880, this work was carried on as required, by the different States bordering the shores of the river, but owing to the unsatisfactory and unscientific results, it was taken out of the hands of the States by the Federal Government. It has now assumed enormous proportions, and annually large sums of money are expended; in fact, up to 1912, $25,000,000 had been spent on improvements of this nature.

Construction Factors. The principal factors to be considered in the construction of levees are: economy of design and maintenance; permanence of the structure; and its suitability to form part of a system when enlargement is considered necessary. This last consideration is of particular importance, since the increased elevation of the river bed resulting from the deposition of sediment, must be met by an increase in the height of the levees; and, where this modification may take place without radical change in the original conception, great saving will result.

The works should therefore follow some definite plan, and, above all, conform to the principles underlying the flow of rivers. Hence, a careful study is necessary, of the river's currents, of the places where erosion occurs, the way in which it is occasioned, the nature of the banks, their height, and the material composing them. Angles should be avoided as much as possible; and where curves are encountered, they should be made as easy as is consistent with good

practice. Generally speaking, the line of a levee follows the river banks, but in the location, regard should be had also for a high and firm foundation, and the line should be selected with that continually in view.

Size and Section. The size and section of a levee will vary with the conditions to be met in that particular locality—that is, specifically, the sort of foundation, the nature of the material of which the levee is to be constructed, and the exposure of the position to tides, winds, and waves.

In America the more usual dimensions are 8 or 10 feet on the top, with a slope of 1 to 3 for ordinary earth, diminishing to proportions of 1 to 5 for sand, with a top width of 15 feet. In foreign countries, the cross-section and mode of construction vary considerably from those in the United States and from one another.

If it is necessary to make the levee more than usually high, what is known as a *banquette* is constructed on the rear face, to reinforce the bank. This is really nothing more nor less than a bench or terrace to increase the width of the lower portion and the stability of the whole. They are usually about 20 feet wide, and have a rear slope somewhat less than that used on the river face.

It is stated that for a cross-section of equal strength in all its parts, side slopes of one to one are all that are required; but in practice this must be modified according to the angle of repose of the material. Steeper slopes than 1 to 3 cause the material on the landward side to slip, while on the river face the material is too easily washed away by the current or the waves.

"Where the exposure to winds is very great, the front slope is often made as flat as 5 to 1, the back slope being then reduced to 2 or 2.5 to 1. It is found that a flat slope is a great protection against the wash of waves, and that a well-sodded *buckshot* levee, with a slope of 5 to 1, will stand a pretty stiff wind. If the sod be once cut through, however, and a hole made in the clay, the latter is liable to be undermined, and the superincumbent masses of earth fall in huge blocks."

The standard section adopted by the Government for all usual conditions, according to Coppee's work, is as follows:

"Crown, 8 feet; front of river slope, 3 to 1; back slope, 3 to 1. Where the levee is over 11 feet in height, a banquette, at an elevation

of 8 feet below the top of the main levee, is added. The slope of the crown of the banquette is 10 to 1; width of crown, 20 feet; and back slope, 4 to 1. Where the foundation is bad or the material weak, the banquette section, and perhaps the front slope of the main levee, are increased."

"The specifications require the levees to be constructed in 2-foot layers, with scrapers, on a well-grubbed and thoroughly-plowed foundation containing a small exploration muck ditch filled back with strong material, the best to be found in the vicinity, and sodded at 2-foot intervals with Bermuda grass."

On that portion of the Mississippi known as the "Fourth District", the dimensions of the standard adopted vary with the height, and are intended to conform more nearly to the supposed theoretically perfect section. These variations may be further modified, if required by abnormal condition of foundation, material of construction, wave wash, etc., as in the other districts.

For levees from 5 feet to 10 feet in height, the crown is 8 feet; the river slope is 3 to 1; and the land slope is 4 to 1 to within 5 feet of the crown; thence to the crown it is $2\frac{1}{2}$ to 1.

For levees from 15 feet to 20 feet in height, the crown is 8 feet; the river slope is 3 to 1; the first 8 feet of the land slope from the ground is 6 to 1; the next 6 feet, 4 to 1; and thence to the crown, $2\frac{1}{2}$ to 1.

In the upper districts 10 per cent of the height, both in wheelbarrow and team work, is required for shrinkage.

These standard sections are expected to withstand the water to within 3 feet of the crown of the levee, without excessive saturation or change of form, and to give unqualified protection under all normal conditions of foundation and materials of construction. When subjected to water above the 3-foot line, though they are intended to remain intact, they cannot be considered standards of excellence, either theoretically or practically.

Height. One of the most important factors to be determined with regard to a levee is the *height.* This needs special consideration, since failure is a foregone conclusion if water is allowed to overtop the crest. It is therefore necessary that all levees be built to a height above that of the highest flood level; and, depending upon circumstances, this will vary from 2 feet to 4 feet above that elevation.

The height is dependent, in a measure, upon the position of the levee, and the direction of the current. For example, an embankment so placed that it lies at an angle to the direction of the current will need to be much higher than one parallel to it, for with the former the water tends to bank up against the levee, while with the latter such is not the case.

Materials. Without taking into account the effect of waves on exposed levees, which necessitates recourse to special slopes and methods of protection, planking, revetments, etc., the whole question of standard section depends on the permeability of the embankment and foundation, that is, the extent of seepage or percolation, and the best form and method for overcoming it in different materials.

In buckshot or clay, which is practically impermeable, the section might be given a strictly theoretical form, dependent alone on the height of the water and the weight of the buckshot, allowing some crown merely for increasing the height in time of excessive flood, the slopes being plane surfaces with an inclination sufficient to insure the required weight to counteract the hydrostatic pressure and the angle of repose of the material.

In cases of permeable materials, light clays, sand, and loam, the levee becomes partly saturated when subjected to high water, the line of demarcation between saturated and dry soil descending in a hydraulic gradient varying in inclination with the soil of which the levee is composed, and being probably very irregular in trace because of the lack of homogeneity of the material in the body of the levee.

In surface soils, subject to direct rainfall or to percolation from adjacent watered areas, the ground water stands at a level dependent on the composition of the soil, both physical and chemical, the natural and artificial voids, and the hydrostatic pressure. In nearly all soils remote from intersecting fissures, wells, or streams, the line or plane of saturation is parallel with the surface of the ground, following the inclination of hill and valley. Where wells, fissures, or river beds occur in the surface soil, the line of moist material, or plane of upper surface of saturation, is inclined toward the fissure, well, or river, the degree of inclination depending on the consistency of the soil.

The power of soils to resist the pressure of water is due to their specific gravity, fineness of comminution, cohesiveness, and the

irregularity of individual particles. Coarse, sharp sand has greater resisting power than that composed of fine, smooth, rounded particles.

The relative strength of materials, as found in this levee district, to resist deformation due to seepage, or their value for levee purposes, is in about the following order: (1) buckshot and gravel tamped in shallow layers; (2) buckshot artificially mixed with sharp sand in shallow layers; (3) buckshot or clay; (4) heavy, strong soil; (5) coarse, sharp sand; (6) light soils; and (7) fine sand, rounded particles.

The material of which a levee is constructed is usually that nearest at hand, but some materials are better adapted to such use than others. Clay is the most satisfactory, as it is not only impervious to water, but also resists wave action well. Sand is not so good, as it is more easily washed away, is less impervious, and requires a greater amount of material, since the slope at which it will stand in a bank is less. Frequently gravel is used as a facing to prevent the washing away of material.

River Bank Protection. While the erection of embankments on the shores of a river serves the purpose of confining the water to the channel itself or to a given cross-section, in no sense does it fulfill the function of preventing the currents from eroding the banks, undermining them, and carrying off the material, until at last failure results.

With this particular object of protection in view, therefore, special works must be undertaken. These will tend to reduce the amount of silt carried by the river and consequently the amount deposited; to preserve a permanent river bed and channel; to protect property, levees, wharves, landings, etc.; and to prevent cut-offs.

Where protection can be given so as to prevent erosion, the flood current will help to remove any deposit that may have collected in the channel, and thus prove to be a helpful rather than a destructive agent.

Forms of Construction. In its simplest form this improvement is secured by placing loose stone along the submerged face of the bank, and by sodding that portion of it which lies above the normal water level, as a precautionary measure during flood stages.

The more complex structures are known as *revetments,* and are divided into two general classes. In the first, the bank for its entire

length is covered by a protective apron of material; while in the second, only portions of it receive this treatment.

All such works consist of two main parts: that below the water level, which performs the function of a foundation; and that above, which rests upon the former and is continually exposed during the dry season. Whatever its nature, the foundation requires care in its construction, since, being hidden from view, it is hard to locate spots where deterioration has commenced. Upon its stability also depends that of the superstructure.

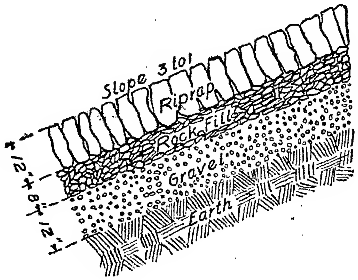


Fig. 9. Simple Form of Bank Protection

Stone. Frequently the method employed will consist simply in depositing stones with more or less care against the bank in question, and continuing this process until, from the bed of the river to a considerable elevation above flood level, it has been completely covered. In case the bottom is soft and the stones settle, it is the practice to dig a trench along the foot of the bank so that the rock may be deposited in this, and serve as a foundation or *toe* for that above.

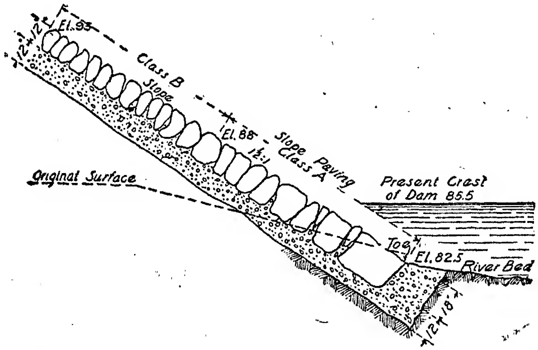


Fig. 10. Bank Protection by Means of Stone and Gravel Where Bottom is Soft

Such a procedure, if properly executed, guarantees stability, and uses less material than in the preceding case, Figs. 9 to 12.

A revetment of dry masonry is sometimes used, and less frequently the stone is laid with mortar; but the advantages gained by

its increased stability are offset by the excessive expense, except, perhaps, where the bank lies within the limits of a city or town.

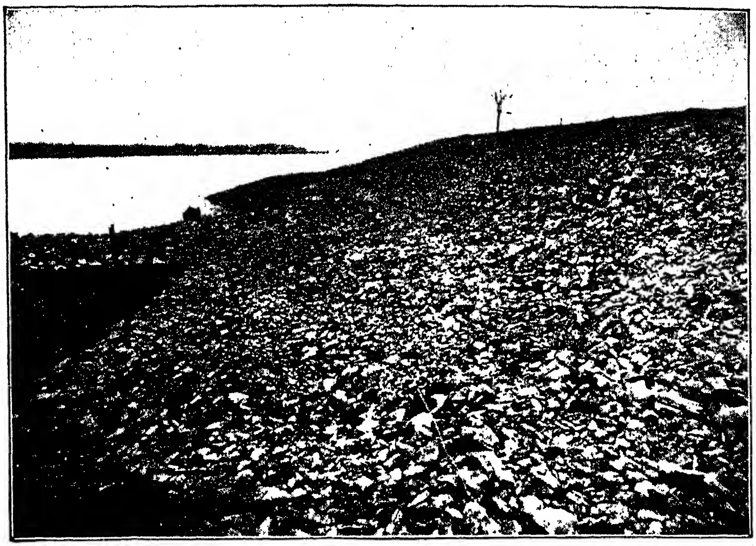


Fig. 11. Bank of Mississippi River Covered with Rubble to Protect It against Erosion

Fig. 12. Bank Protection on the Mississippi River
Placing the broken stone in position. Partially sunken mattress shown in foreground.

Dry masonry consisting of either field stone or roughly cut flat stone placed by hand, is frequently laid on a foundation of gravel

or sand, the sand or gravel adapting itself to the irregularities of the bottom of each stone and thus securing an even upper surface. This method is often employed on canals where the banks of the prism need some sort of protection, and where erosion would otherwise occur as a result of the wash of small steam craft or the rubbing of tow ropes.

Piling. Round or sheet piling is also frequently employed. It has the advantages that even less material is needed and that it is more stable. It is subject to decay as all wood is, if alternately wet and dry; for this reason, therefore, all timber should be carefully inspected during the dry season.

Fig. 13. Mooring a Mattress to the Bank of the Mississippi River
The mattress is subsequently sunk by loading stones upon it, and makes an effective foundation for jetty construction in the improvement and control of the river.

Fascines and Mattresses. In this country and abroad extensive works of defense have been constructed of brush, saplings, poles, etc. This method has the particular advantage of being cheap, and of using material usually close at hand; besides which, it gives very satisfactory results. The names applied to the material so employed are *fascines* and *mattresses;* the former consist of round bundles of long willow or branches of other sorts of trees tied together; while the latter consist of branches or poles woven together in horizontal layers.

In Holland, the fascines used vary in length from 8 feet to 13 feet, and from $1\frac{1}{3}$ feet to $1\frac{2}{3}$ feet in diameter; while in Germany they are 13 feet to 16 feet in length, and from 1 foot 2 inches to 1 foot 10 inches in diameter.

Mattresses are formed into various shapes; but they are usually horizontal layers of brush, woven together by poles, wire, timber, etc. Iron wire has been found to rust readily, so that galvanized wire or wood has been substituted, with very satisfactory results.

The use of mattresses has been developed to a very great extent along the Mississippi, where cottonwood and willow are abundant and stone usually conveniently located. In this region, it is the method pre-eminently in favor for protecting the banks from scour; and the practice is here followed and developed to a greater extent than anywhere else in the world, Figs. 13 to 18.

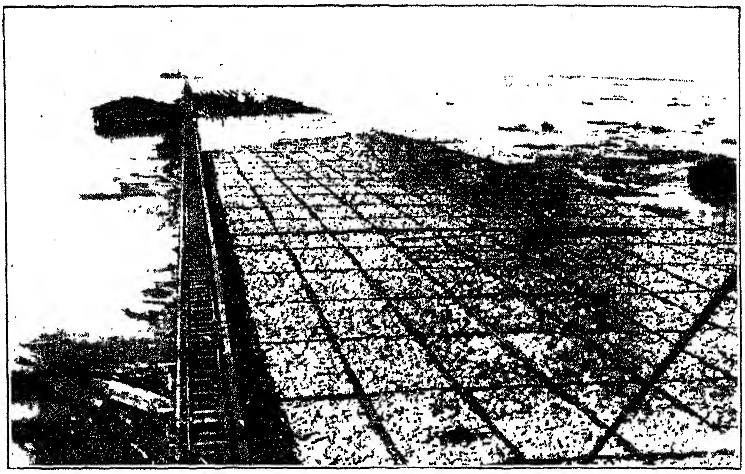


Fig. 14. One of the Huge Mattresses Used in Jetty Construction on the Mississippi River
Mattress 100 feet wide, in position in shallow water, covered with the broken stone used to sink it. The cross-sections are produced by the scantlings used to tie the mattress together.

A mattress 300 feet wide by 1200 feet long represents a superficial area of about 8 acres; and when one realizes that this vast willow carpet, over a foot thick, is placed on the bottom of the river at depths of from 40 feet to 100 feet, and against currents of from 5 feet to 8 feet per second, the difficulty of the enterprise will be appreciated. Though much of the revetment from Cairo to New Orleans has needed repairs from year to year, and in some reaches has required renewal as a whole, it may be said to have been eminently successful in the protection of cut-offs and outlets, and fairly so in the control of bank caving and the resulting change in position and flow of the river.

At some points where the material of the bank was friable and the currents very strong, the earlier forms of revetments proved too light and were entirely swept away, the shore line continuing to move back. Also, considerable reaches of protection work needing repairs and reinforcements at the ends, have been destroyed because of the lack of funds. But in the later work the results have been beneficial and satisfactory, and the loss but slight.

The method of constructing these revetments as practiced on the Mississippi River is as follows: The bank is cleared 50 feet back from the top, and graded on a slope of 1 to 4 from the low-water line.

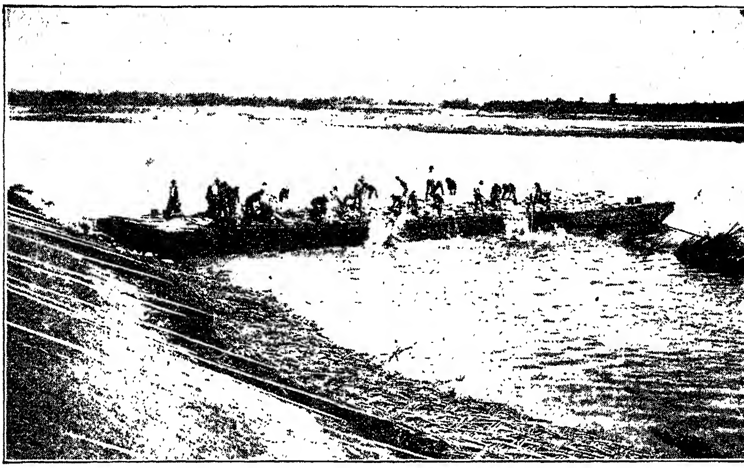


Fig. 15. Sinking a Mattress by Throwing Stones upon it from a Scow
The mattress is to serve as the foundation of a jetty for river-bank protection on the Mississippi.

This slope is then dressed by filling the holes and removing tree stumps, etc. At the upper end of the section to be protected, a cluster of piles called an *abutment* is driven; and below them, along the shore, single piles, spaced 100 feet apart, are driven to hold the mattress in position for the full length of bank to be protected. The mattresses are built on barges, floated to position, and then sunk by placing rock on top.

Constructing Mattresses. The following description, based on the Government report, illustrates the method of constructing the mattresses themselves: hardwood poles, as large as can be conveniently handled by a gang of men, and reasonably straight, are laid in

two lines on ways over and parallel to the inner gunwale. These poles lap each other between 10 and 15 feet, the two lines breaking joints. Where they lap, they are spiked together, and they are also tied together by No. 12 galvanized wire at intervals of 10 feet. Two ties are made at the laps. This line of poles is as long as the mattress is wide. About 7 feet 6 inches apart on these poles, and at right angles to them, the butt ends of weaving poles made of live willow or cottonwood brush from 4 inches to 6 inches in diameter and 25 feet to 30 feet long are fastened with spikes and wire. Another set of poles similar and parallel to the first is placed on these, and securely spiked and wired. To facilitate weaving, the top and the bottom of the weaving poles are shaved and the knots trimmed. A cable made of 8 strands of No. 12 wire is fastened around the head of the

Fig. 16. Building a Mattress from a Trestle Where Water Was Too Shallow to Allow of Bringing in a Mattress Which Had Been Built on the Regular Mattress Building Ways

mat at every third weaving pole and run up alongside of it, the end being fastened thereto by two staples.

These cables are 24 feet long, with an eye in one end, to which, after each shift of the mat, a new length is looped in weaving. The continuous cables are thus formed in the mat, greatly strengthening it longitudinally. When this head is finished, lines are connected to it from the shore, passing under the mooring barges.

The brush used for weaving is live, straight willow of any length over 25 feet, and between 2 inches and 4 inches thick at the butt.

The butts are placed over one weaving pole, and project 2 feet beyond, being woven at the other end over the next pole, under the third, over the fourth, and so on, the light ends being always left on top. A strip 5 feet wide is thus woven. In the next strip, the

butts are reversed, the butts changing direction every 5 feet. When the mattress is woven within 2 feet of the end of the poles, giving about 22 feet length of mattress, it is swung into position with the accompanying barges. The head lines on the barges and mattress are slackened until the barges are nearly normal to the shore, with their inside edge resting against the pile abutment. The slack in the mooring cables is now taken in from the bank, and the strain equalized. They are then fastened permanently with clamps, as are also the mattress head lines. An entire shift, 22 feet long, is then launched, a new set of weaving poles being spliced to the projecting ends of the first set. This is continued as described, to within 2 feet of the top of

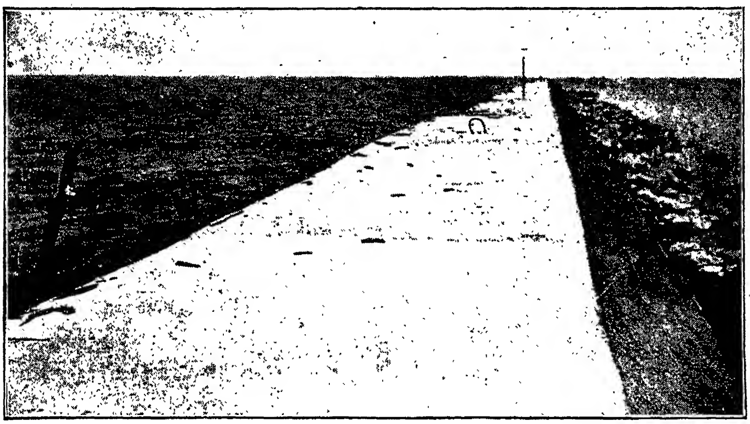


Fig. 17. A Completed Jetty on the Mississippi River
A concrete top resting on a foundation of stone-laden mattresses.

the second set of poles, when another launch is made; and so on until the full length of the mattress is obtained.

When three shifts have been launched, the construction of a top grillage or framework is begun. This consists of a line of poles laid over and parallel with the weaving poles, lapping each other, butts to tops, between 6 and 8 feet, and wired to the weaving poles every 4 feet by lashings 2 feet long, made of 2 strands of No. 10 wire; transverse poles 8 feet apart for the first 100 feet, and thereafter 16 feet apart, are placed in similar manner, and fastened to the longitudinal ones at the intersections by 2-foot lashings made of 4 strands of No. 12 wire. The purpose of the grillage is to make pens in which the stone is retained, so as to strengthen the construction.

The first set of transverse poles along the inner edge are hardwood, set 8 feet apart throughout the length of the mat, and are used to connect to the river mat the shore mat, which is also being built.

This shore mat is constructed of hardwood poles of the size of the weaving poles, lashed and spiked to the river mattress, with willow or cottonwood poles spliced to these until they reach up the slope about 40 feet. Alongside, and fastened to each of the hardwood poles, is a cable of 8 strands of No. 10 wire, one end of which is fastened to the adjacent weaving poles, and the other to the willow poles extended on the slope. Upon the transverse poles are laid longitudinally willow or cottonwood poles 8 feet apart, beginning with the first set about 4

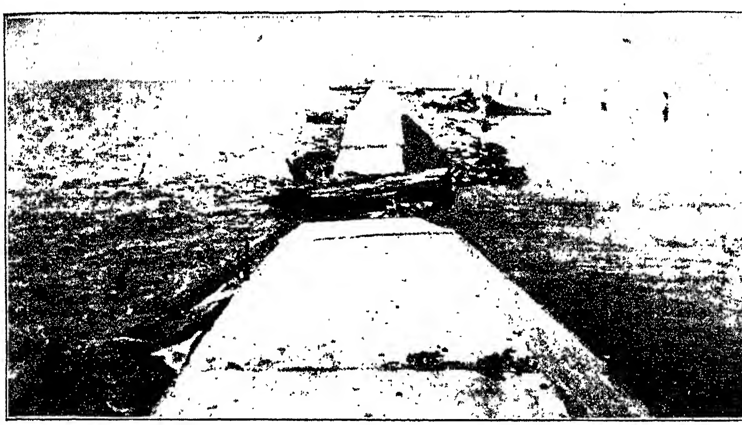


Fig. 18. A Break in a Mississippi River Jetty
The concrete wall capping the jetty has sunk, together with its foundations, several feet into the soft mud bottom on which the jetties rest. Fortunately such breaks are not of frequent occurrence.

feet from the edge of the mat. The latter poles are wired to the former at their intersections. The longitudinal poles are carried on lines 8 feet apart, up to the top of the slope; and on their lower side, 8 feet apart, are driven stakes 2 feet 6 inches above ground, to the tops of which is loosely fastened a lashing wire whose bight has been passed under the pole. These stakes are used down to the pole nearest the water edge. Upon this framework is laid willow brush diagonally, with the butts toward the top of the slope, and breaking joints throughout. A second layer of brush is put on in the opposite direction, the two thus being at right angles to each other. On top of these layers a second pole framework, fastened similarly to the

first, is placed and fastened down firmly by the lashings mentioned above as being tied to the stakes.

As fast as the river work and shore work are finished, transverse cables are run across the entire width of the mat at 16-foot intervals, carried to top of the bank, hauled taut, and fastened to trees, stumps, or "deadmen" placed for the purpose. These are bound to the mat every 16 feet with lashings. When 400 or 500 feet of river mattress have been completed, longitudinal cables are run out from the mooring barges and securely attached to the mat at 16-foot intervals. One of these is placed close to the edge of the mat, the others at 30, 37, 38, and 42 feet, respectively. The mats are ballasted and sunk by placing stones on them, and, when sufficiently submerged, the scows are floated over them, and the material dumped from them. Owing to the undermining of the outer mattresses, these are generally more firmly built, and consist of bundles of fascines bound together.

Another method of protection similar to the above is by means of a spur revetment, where the mattresses are built out into the stream at right angles to the shore. These are placed at intervals along the bank that is threatened, and seem satisfactorily to check the destructive effects of the current.

Dikes. Dikes are a form of structure "for guiding a stream along a caving bank and protecting the same from undermining, for confining and directing the water at bars and shoals, and for closing secondary arms of a river," Fig. 19.

There are three general forms: *spur dikes; longitudinal dikes;* and *submerged spurs*—the first two being frequently used in combination.

Spur Dike. Spur dikes are used a great deal in Germany, but only to a small extent in the United States. Their object is to improve navigation, and this is accomplished by placing them along the shore at intervals. They run out into the stream at a small angle with the shore, and thus tend to confine the current to a smaller cross-section. Their form of construction causes the deposit of silt between the spurs, which material tends to consolidate and help protect the dikes themselves from floods and ice. They thus form a new bank which is continuous between the heads of the dikes. This system generally requires more than one to produce any increase in the depth of the channel, as the water is confined only at the outer end, or "head", of the dike.

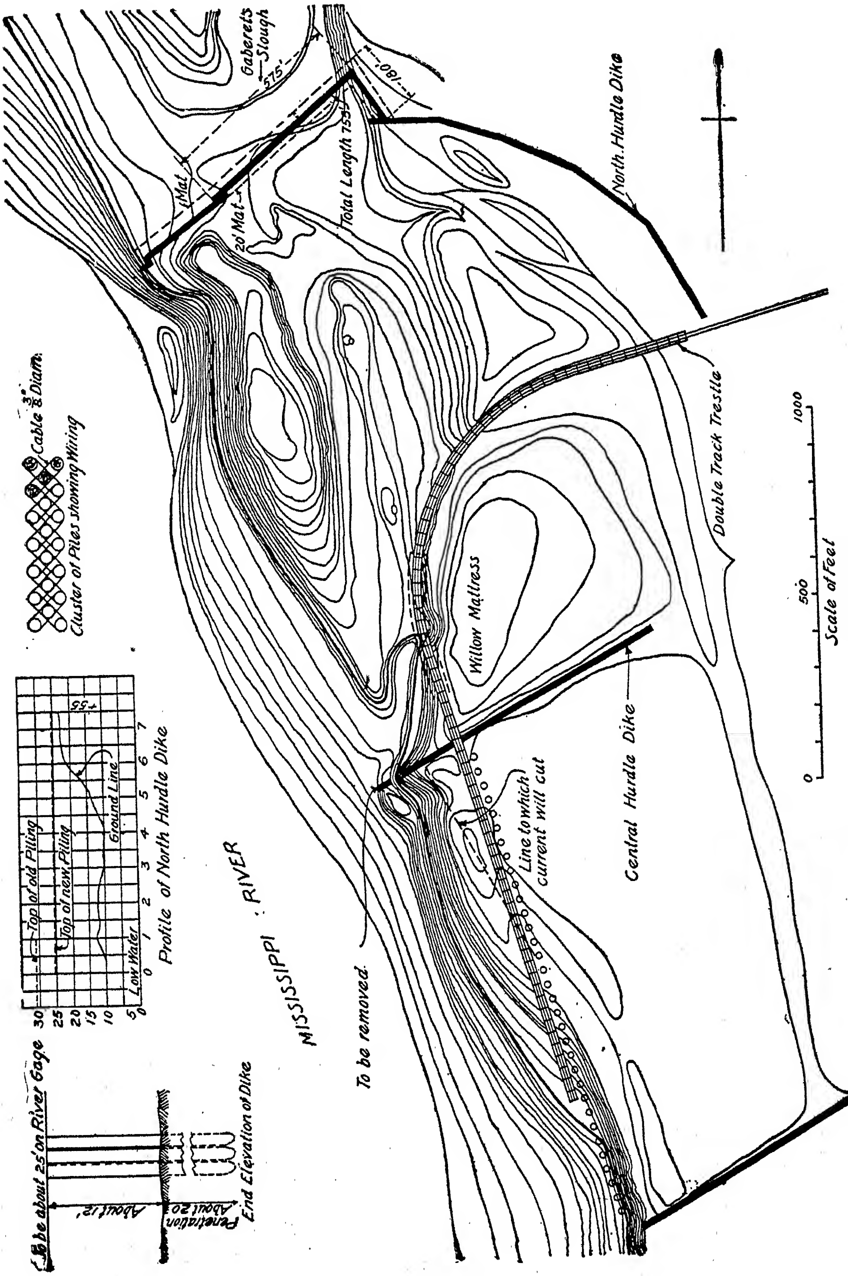


Fig. 19. Protection of a Trestle by Means of Dikes

Longitudinal Dike. Longitudinal dikes are made parallel to the shore or the current, their object being the same as in the case of spur dikes. They are used on the Ohio to some extent, where they are built of wooden cribs filled with stone. Generally they begin at the bank and extend downstream, with a gentle curve away from the shore, when they run parallel to it for some distance. Behind them, as in the case of the spur dikes, the space fills with silt and mud, and thus tends to form a new bank. Such construction is highly advantageous where it is desired to scour out bars, and thus secure greater depth in the channel at low water. During normal flow, the tops are submerged. Occasionally they become undermined by the swift current produced; besides this they are expensive to maintain, and need constant attention.

Submerged Spur. Submerged spurs are built in the bottom of a stream or a river, to correct the influence of scour produced by one or the other of the two forms of dike already described. Their object is to produce a silting-up of the bottom, or to prevent the bottom from being scoured out below a certain depth, a situation that would endanger the bank or other protective works.

EROSION AND TRANSPORTATION

Where the flow of water in a river channel is uniform, and the channel comparatively straight and regular, there is little tendency for the stream to erode the banks, but where bends and curves occur, this tendency does exist and sometimes to a marked degree. Under such circumstances, the greatest depth is near the outer portion of the bend, while the shallow portion is opposite. This is due to the centrifugal force of the current, which resists the change in direction and scours away the outer bank.

Current Action at River Bends. Perhaps the most complete explanation of what occurs in the stream, when the water encounters a bend or curve in the river, is to be found in the *U. S. Geological Survey Report* on the subject, issued in 1914: "The Transportation of Débris by Running Water".

In a straight channel the current is swifter near the middle than near the sides, and is swifter above mid-depth than below. On arriving at a bend the whole stream resists change of course, but the resistance is more effective for the swifter parts of the stream than for the slower. The upper central part is deflected least and projects itself against the outer bank. In so doing

it displaces the slow-flowing water previously near that bank and that water descends obliquely. The descending water displaces in turn the slow-flowing lower water, which is crowded toward the inner bank, while the water previously near that bank moves toward the middle as an upper layer. One general result is a twisting movement, the upper parts of the current tending toward the outer bank and the lower toward the inner. Another result is that the swiftest current is no longer medial, but is near the outer or concave bank. Connected with these two is a gradation of velocities across the bottom, the greater velocities being near the outer bank. The bed velocities near the outer bank are not only much greater than those near the inner bank, but they are greater than any bed velocities in a relatively straight part of the stream. They have therefore greater capacity for traction, and by increasing the tractional load they erode until an equilibrium is attained. On the other hand, the currents which, crossing the bed obliquely, approach the inner bank are slackening currents, and they deposit what they can no longer carry. It results that the cross-section on a curve has the greatest depth near the outer bank.

Relation of Velocity to Transporting Power. This erosion, or wearing down of the river bed or banks, is continually going on when deposition of suspended matter does not take place, but when the river becomes saturated, so to speak, with transported material and can handle no more, some of the particles, and naturally the heaviest, will be dropped to the bottom.

The greater the velocities, the greater the transporting and erosive power of the stream. As these higher velocities exist in the mountainous districts, where the slopes are greatest, here the larger material will be eroded or moved; and where the velocities are reduced in the flatter portions of the stream the greatest amount of material will be deposited, until finally, where the current is very slow, only the finest silt can be carried.

In these latter localities the material carried by the swifter current will be deposited in the bed of the stream or along the shores where the velocities are low. In this manner a river will modify its profile and may actually build up its banks in times of flood. This general process is called *gradation;* a river that builds up its bed is said to *aggrade,* and one that wears down its bed is said to *degrade.*

Streams may also be spoken of as *corroding,* or *rock-walled,* or *alluvial,* but it must be remembered that no stream flows through a country of the same characteristics from the source to mouth, and in consequence most streams may be designated by at least two and frequently three of the above terms.

TABLE IV

Transporting Power of Water for Various Materials

MATERIAL	VELOCITY (ft. per sec.)
Potter's clay	0.26
Sand deposited by clay	0.54
Large angular sand	0.71
Gravel—pea size	0.53
Gravel—bean size	1.07
Round pebbles—thumb size	2.13
Angular flint stones—hen's egg size	3.20

When the slope is pronounced, the velocity, being appreciable, will produce under usual conditions, at least where the nature of the stream is alluvial, a straight watercourse, but where the slope is gentle the channel will be found to meander, and to have many curves and turns.

Table IV shows the velocities of water necessary to the transportation of various materials.

The diameter of a body that may be moved by a current varies as the square of the velocity; and the weight of such bodies, as the 6th power of the velocity.

The force exerted in moving sand or pebbles by water is proportional to the square of the velocity and to the area exposed. Thus the force required to move a body of diameter d will be

$$F = cd^2v^2$$

in which c equals some constant. If motion just occurs, then F is proportional to the weight of the body also, because the frictional resistance of one body upon another varies as the normal pressure or weight.

The weight of a sphere varies as the cube of the diameter, so that

$$d^3 = cd^2v^2$$

or,

$$d = cv^2$$

and, since d varies as v^2, the weight, which is proportional to the cube of the diameter, must vary as v^6. Consequently, an increase in velocity causes greater increase in transporting capacity.

As the weight of such material in water is about $\frac{1}{2}$ its weight in air, the frictional resistance to motion is slight; hence the pebbles, etc., are easily moved by moderate velocities. It may be said, there-

fore, that ordinarily small material such as loose earth will be carried or rolled along the bottom by velocities of 2 feet per second or less.

Character of a River. The general character of a river may be said to depend upon the united action of three factors: *discharge,* which is variable periodically with the changes of the seasons, and from year to year, and which also undergoes permanent changes as the result of deforestation of the watershed and other conditions produced by the work of man; *magnitude of slope,* which is likewise variable; and *nature of soil,* which is variable in different localities.

For a river to be navigable, it is necessary that it should have a sufficiently deep channel throughout its entire length. A river may have much water; but if the fall is considerable and the soil unstable, it cannot have a deep channel. On the other hand, there are rivers with a relatively small discharge and great fall, which are yet quite suitable for navigation, owing to their hard bottom, which effectively resists erosion and transportation of its material, and thus preserves a relatively permanent river bed.

Shoals are portions of the river bed which resist erosion and therefore produce shallow spots.

Bars, on the other hand, are formed by the deposition of transported material, either from the natural conditions of the current or from the artificial ones produced by the construction of piers or breakwaters. The effects of such constructions vary, of course, with the nature of the material of which the river banks and bed are composed, but more particularly with the angle at which the constructions are laid relative to the axis of the stream.

When river banks are protected, the silt carried down during the flood is finally deposited, on the lowering of the stream, in the bed of the channel, rather than over the adjoining lands as would be the case were it unconfined. In consequence of this, and by successive depositions during long periods of time, the bed of the stream becomes elevated above its former level, so that the banks must be raised at the same time to keep pace with the silting-up of the stream.

In Japan there are cases where this process has been in continuance for so many years that the river now flows at a considerable elevation above that of the surrounding country, making it preferable to carry railway lines beneath the stream by tunnels, rather than over them by bridges. This is the extreme condition, of course;

but observations show that the same thing is going on along the course of the lower Mississippi River.

Improvement of the Lower Mississippi. As an illustration of the practical application of the foregoing principles, let us refer to the notable work of Captain James B. Eads, in connection with the improvement of the lower Mississippi River. This river, the largest in the United States, indeed one of the largest in the world, has furnished a highway for water transportation ever since the American continent was discovered, and, in the days before the introduction of the steam railroad, even more than at present, was a very big factor in furnishing a cheap means of carrying goods from one section of the country to another.

As early as 1726 efforts had been made to improve the outlets of the Mississippi, but none were successful as none of the schemes provided for the transportation of the material in suspension, and this was what was needed. At New Orleans the river was then ½ mile wide and 150 feet deep, but where it entered the salt water in the Gulf of Mexico, it provided only very shallow channels. As is natural in all large rivers, it carried considerable suspended material, and, upon entering the salt waters of the Gulf, deposited this, forming a delta 107 miles below the city of New Orleans.

This delta is provided with three principal main channels, called the Southwest Pass, the South Pass, and the East Pass. The Southwest Pass in 1874 had a depth of 18 feet and a sectional area of 60,000 square feet; the South Pass, a depth of 8 feet with a sectional area of 24,000 square feet; and the East Pass, a depth of 11 feet with a sectional area of 69,000 square feet. It will be seen from this record to what an extent the channel has been reduced in passing from New Orleans to the Gulf.

Eads' Accomplishment. In 1874, Captain Eads proposed to correct conditions so as to make it possible for any ocean-going vessel to reach the port of New Orleans, and with that end in view suggested that the Government permit him to undertake the work of deepening the channel of Southwest Pass, where the depth of the water was greatest, with the understanding that if his methods did not prove successful he should receive no compensation for the work. This the Government refused to do, but Congress did authorize him to put his plan into effect with respect to the small South Pass.

Captain Eads' plan was based upon accurate knowledge and the application of the hydraulic principles, diversion, concentration, and regulation; that is, he arranged for as much of the water as could be controlled to pass through South Pass, so that at the period of flood, when over $1\frac{1}{4}$ million cubic feet of water per second are delivered through the delta, the scour would be so great as to deepen the channel. This was accomplished by building jetties parallel to the banks in order to confine the water within the channel, closing all crevasses or openings leading off from the main stream by dams so that no water could escape, and then regulating the flow of the river that it might accomplish what otherwise could not be done.

Work began on this undertaking in June, 1875, the line of the jetties being based on an accurate survey, complete and exhaustive data as to the characteristics of the river being at hand, and the principles of Hydraulics being applied in an intelligent manner. That the work was a complete success is shown by the fact that in August, 1876, the channel in the South Pass had deepened from 8 to 20 feet; in February, 1878, it had reached a depth of 22 feet; in July, 1878, a depth of 24 feet, and in October, 1878, a depth of 30 feet. Furthermore, this improvement has been permanent, for not only has the depth been maintained—it has been increased. No more satisfactory demonstration of river improvement can be presented in the entire realm of Civil Engineering.

Similar improvements have been undertaken elsewhere on these same lines and with equal success; thus at Tampico, Mexico, by means of jetties the channel depth of Panuco River, where it enters the Gulf of Mexico, within a period of two years from the beginning of construction, was increased from 8 feet to $19\frac{1}{2}$ feet, which three years later had increased to 24 feet.

HARBORS

PROBLEMS OF HARBOR IMPROVEMENT

A *harbor* may be said to be "an indentation or inlet on the shore of a sea or a lake, so protected from winds and waves, whether by natural conformation of the land or by artificial works, as to form a secure roadstead for ships"—in other words, a place of refuge or of safety. But the term has a broader signification which includes

all those works required or used to facilitate the repairing, loading, or unloading of ships. Thus the subject of "Harbors" embraces questions of anchorage, piers, docks, buoys, lighthouses, etc.; while "Harbor Improvement" concerns itself with the betterment of them.

In a few places natural harbors exist with ample depth of water to accomodate the largest ships—such as at Rio de Janeiro, New York, San Francisco, and Portsmouth, England; but frequently, on the other hand, it will be found necessary to construct sheltered sites along an exposed seacoast, both for trade and for refuge in time of storm. It will be found that, generally speaking, harbors furnishing natural shelter are formed by bays, creeks, or projecting headlands, although they may require one or more breakwaters to protect the roadstead from the sea.

No general rules may be laid down by which the engineer may hope to design or improve a harbor, but rather must he employ the material at hand, and utilize the natural conditions to the best of his ability, to obtain the end in view, as no two localities will be found to present the same problems.

Points for Preliminary Investigation. In order to undertake the work intelligently, therefore, the engineer should make a careful survey or examination of the locality, so that complete data may be at hand from which he may develop his plans. Such an investigation would include:

(a) A comprehensive and detailed survey, the limits of which can be best decided after an inspection has been made of the site and its general surroundings. Soundings should be taken at intervals sufficiently short to admit of contours being accurately laid down. These should be plotted with special care over rocky ground. The nature of the sea bed should be ascertained by securing a sample of the bottom at each sounding, and, when the probable site of the works has been tentatively decided upon, borings may, if necessary, be taken in order to obtain more reliable information than the surface samples afford. Careful observations should also be made on the tides; current; height and direction of the heaviest waves; description, extent, and direction of littoral drift; direction and force of prevailing winds; fetch; and so forth.

(b) A careful examination of the district, with the view to ascertaining what stone, sand, gravel, and other materials—not forgetting water—are available for the works.

(c) A consideration of possible sites for workyard and quarry, and of the approaches or means of access thereto; also of the means of transporting stone and other materials to the works, and possibly in some cases, of landing not only materials, but also heavy "plant".

(d) A careful consideration of the general facilities which the locality affords for the construction of the works contemplated, including the important question of labor.

Types of Harbors. All harbors may be divided into two general classes: (1) those designed to serve *commerce;* and (2) those constructed as places of *refuge.* These may each be again subdivided into *natural* and *artificial* harbors.

Commercial Harbors. These are chiefly for the purpose of loading and unloading the freight carried by ships and may consist of almost any arrangement of piers and breakwaters to enclose and tranquilize the water, together with the quays, wharves, docks, etc., which are maintained in conjunction with them. Frequently on exposed coasts, what is known as a *compound* or *double* harbor will be constructed, consisting of outer protective works, with an inner basin for the mooring of ships, Fig. 20.

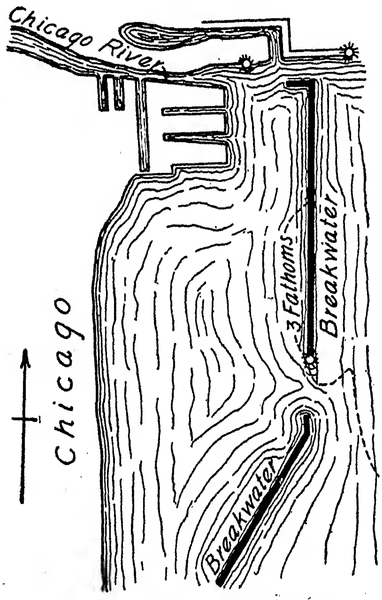


Fig. 20. Harbor of Chicago, Illinois

Harbors of Refuge. The term *harbor of refuge* may be applied in both a generic and a specific sense. Generally speaking, it refers to a port in which craft may obtain shelter, no matter what the conditions of the weather, and which is located on a portion of the coast much frequented, but where no protection has been supplied by Nature, Figs. 21 and 22.

In this sense, a harbor of refuge should meet the following requirements: (1) be easy of approach; (2) afford secure anchorage at various depths for all sorts of vessels; (3) possess facilities for furnishing supplies and making small repairs; and (4) have natural capabilities for the construction of works of defense at a small cost.

On the other hand, in a special sense, a harbor of refuge is one which is located primarily with a view to its strategical possibilities,

and rather as a refuge for battleships, both from the enemy and the weather. Such a harbor would naturally be a national undertaking,

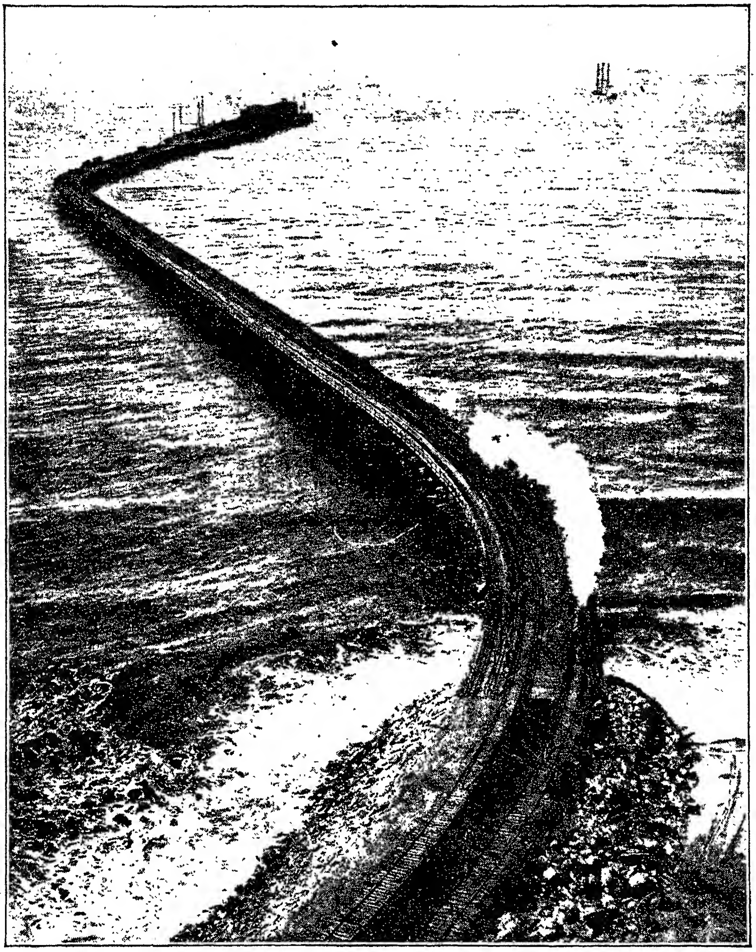


Fig. 21. Improvement of San Pedro Harbor, Port of Los Angeles, California

View showing about 4700 feet of the trestle used in building the great 9000-foot breakwater for which Congress appropriated $3,000,000. Beyond the end of the trestle here shown the completed structure makes a turn to the left, creating, on the left, an outer harbor of 550 acres, safe at all times. The breakwater is of the "mound" type, 190 feet wide at base, tapering to 20 feet wide across the top, and consists of large rubble topped with dressed stone laid regularly without mortar. The inner harbor is to cover about 1200 acres, with a minimum depth of 25 feet.

and should possess the following characteristics: (1) be centrally located with regard to the defensive works and the shipping of the coast at that particular point; and be situated where the enemy can

be conveniently observed; (2) afford a greater anchorage and depth than in the case of the general harbor of refuge; and (3) be connected with the great railway systems of the country, so that troops, coal, supplies, repairs, etc., can be readily transported there. (See Fig. 23.)

Effects of Natural Forces. It seldom happens that a harbor will be found so satisfactory that no improvements whatsoever need be made. But while this, in some cases, will be but a simple matter, in others it may mean practically the entire reconstruction of the port. Thus, for example, the Harbor of New York, though one of the

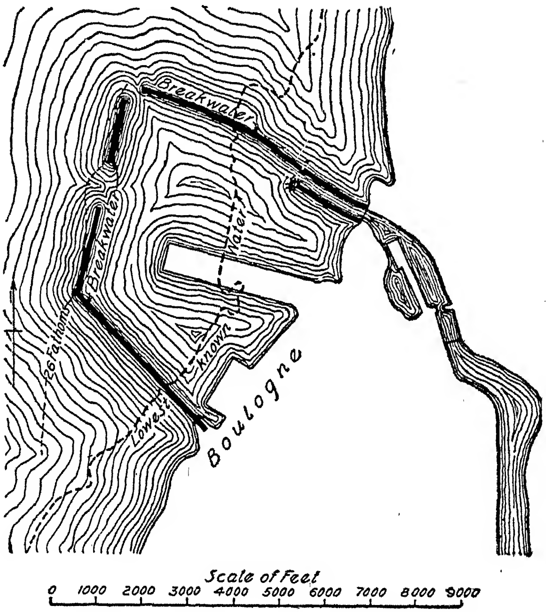


Fig. 22. Harbor of Boulogne, France

finest natural harbors in the world, needs constant attention in the matter of dredging to prevent the channel from becoming silted-up and thus precluding the entrance of heavy draft vessels to the Upper Bay and the docks of the city.

As a general rule, most harbors are found at the mouth of a river, and yet the improvement of a river mouth for harbor purposes is one of the most difficult undertakings in harbor engineering. Usually, the mouth prevents easy entry because of the bars or shoals formed there by the sudden deposition of material carried in suspension by

the river; for, when the current strikes the larger body of water, its velocity is at once checked, and the silt sinks to the bottom.

Waves also tend to increase this action, not alone because they retard the current of the river, but because they carry the material

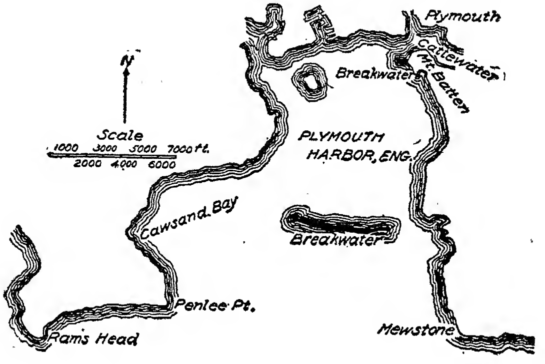


Fig. 23. Harbor of Plymouth, England

back to the beach line itself. Cases occur where prevailing winds blowing toward shore, in combination with long periods of drought, will cause the waves to deposit silt to such an extent as to block the channel completely. Where the mouth is wide, it usually happens that more than one bar—sometimes many bars—with corresponding channels will be formed in this manner.

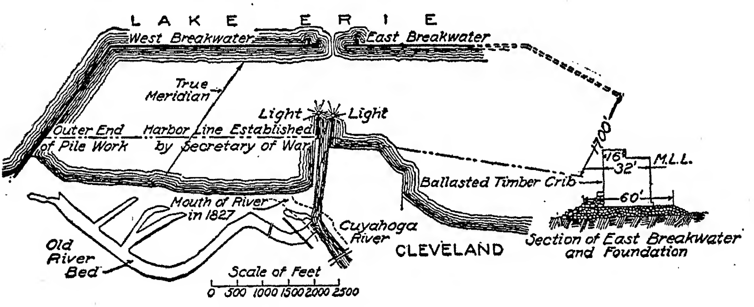


Fig. 24. Harbor of Cleveland, Ohio

It should therefore be the purpose of the harbor engineer to use the forces of Nature as much as possible in the design of such works, whether these be protective in character or not. This will consist in:

(1) Directing the currents so that they will tend to scour out and deepen the channel rather than silt it up. In this connection, the transporting power of the ebb tide should be taken into consideration, so that the material in suspension may be carried out to sea rather than back into the harbor at flood tide.

(2) Increasing the amount of water flowing over the bar at ebb tide, by enlarging the harbor through dredging, and preventing any obstruction to the flow through the agency of curves. It has been proved that reclamation or other works reducing the capacity of the tidal compartment, and consequently the quantity of tidal water passing through a harbor entrance, injuriously affect the depth of water at the entrance in proportion to the amount of water excluded, the converse being equally true.

(3) Protection of the outgoing current from being deflected or retarded by prevailing seas. This is frequently done by means of breakwaters.

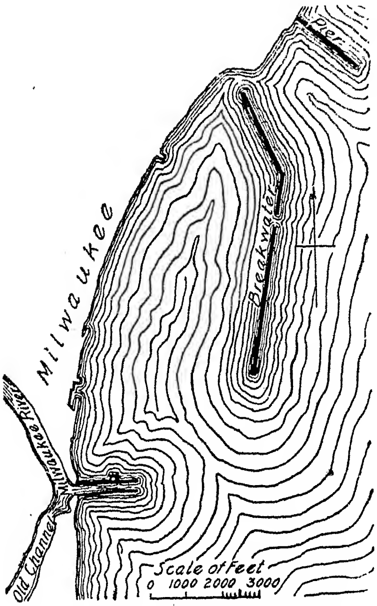


Fig. 25. Harbor of Milwaukee, Wisconsin

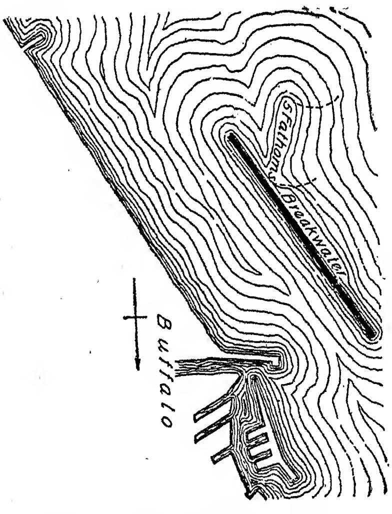


Fig. 26. Harbor of Buffalo, New York

Such harbors as indicated will almost always require periodic dredging, and generally the presence of breakwaters or training banks, Figs. 24 to 26.

It has been observed that when fresh water meets salt, as in the case of a river flowing into the ocean, the former tends to flow over the latter. It is evident, therefore, that, while the tide is at flood, the river will flow on top of the ocean water and thus perform no service in the function of scouring. In consequence, the ebb tide alone must be depended upon to bear away the material deposited.

Winds. Winds cause the waves; and the latter, the destruction of protective works. There are certain periods of the year, however, at which storms are more frequent and severe than at others, and during such times more than ordinary care must be exercised to guard against their

TABLE V

Pressures Corresponding to Wind Velocities

Velocity (mi. per hr.)	Pressure (lb. per sq. ft.)	Velocity (mi. per hr.)	Pressure (lb. per sq. ft.)
10	1.0	60	18.0
20	2.5	70	25.0
30	5.0	80	32.5
40	8.0	90	41.0
50	13.0	100	50.0

destructive action. The time varies in different localities. The strength, duration, frequency, and sequence of winds are matters of vital importance to be determined and studied, if protective works are to perform their useful function.

Relative Pressures and Velocities. Apparently no exact relation exists between the velocity of the wind and the pressure it exerts, as the latter seems to vary with the size and form of structure against which the wind acts; but Table V shows the pressures that are quite generally used for various velocities, in the design of structures that must withstand such action.

TABLE VI

Resistance Offered to Wind

(Bodies having same cross-section but various shapes)

Body and Arrangement	Comparative Resistance
Cone (vertex toward wind)	126
Cone (base toward wind)	291
Cylinder (end toward wind)	285
Hemisphere (flat side toward wind)	288
Hemisphere (round side toward wind)	119
Sphere	124

Relation of Wind Pressures to Lighthouse Shapes. Table VI shows quite clearly that the wind pressure upon an object does not depend only upon the area of the surface that the object presents to the wind, but also upon the form of the object; and it may be added, that its inclination to the direction of the wind will have a most powerful effect upon the resultant force acting against it.

From this fact is is evident that the pressure is greatly reduced by presenting curved surfaces rather than flat surfaces to the wind;

and for this reason, it should be noted, nearly all lighthouses subjected to gales are constructed in the form of a cylinder.

Beaufort's Scale of Intensity. Table VII, known as Beaufort's scale, has been adopted to indicate or classify the degree or intensity of a storm in terms of the miles per hour at which the wind blows.

TABLE VII

Terms of Storm Intensity

(Beaufort's Scale)

Description	Velocity (mi. per hr.)	Description	Velocity (mi. per hr.)
Air, light	8	Gale, moderate	40
Breeze, light	13	Gale, fresh	48
Breeze, gentle	18	Gale, strong	56
Breeze, moderate	23	Gale, whole	65
Breeze, fresh	28	Gale, storm	75
Breeze, strong	34	Hurricane	90

Records are on hand to show that the wind does sometimes reach "hurricane" velocities, and even 95 and 100 miles per hour have been known, but these are extremely exceptional.

Wave Effects. Naturally, it has been attempted to establish a relation between the velocity of the wind and the height of the waves, but it may be freely acknowledged that no reliable data showing a connection between the two has been obtained.

It should be pointed out that winds affect harbors in various ways besides that of causing waves to do damage. Thus the direction of prevailing gales must be determined so that the harbor entrance may be planned to avoid them, as otherwise it might prove difficult or impossible for boats to enter during a storm. Furthermore, if the entrance is poorly designed, it may lead the heavy seas into the harbor and cause a dangerous "run". Vessels themselves are less affected by wind in these days of steam craft than when all vessels carried sail, so that from this standpoint the subject need receive little consideration.

Waves. While the theory of wind waves is too abstruse to be discussed fully in this treatise, it is a fact that a knowledge of wave action and wave intensity is absolutely necessary to the proper design and construction of sea walls, piers, breakwaters, etc.

It is a mistaken notion that the water in a wave travels, as may be readily demonstrated by observing the movement of a chip floating on the surface. Instead of passing onward with the waves, it will be found to oscillate between certain points, the distance between which bears a fixed relation to the length of the waves or the distance from crest to crest.

When waves pass into shallow water, their character changes for the worse as the friction along the bottom retards the lower particles and in a diminishing degree toward the surface until the crest overturns and *breaks*. This generally occurs when the water has a depth equal to the height of the wave at which stage a wave does its most destructive work. This, then, is a very important consideration in connection with breakwaters having a long foreshore formed of rubble or other material as is the case at Plymouth, England, but it may be considerably neutralized by carrying the rubble high enough so that the waves will break on top of the superstructure, when the force will be largely spent.

Thus the length of waves diminishes as they approach shore (i.e., the distance from crest to crest diminishes); and their velocities also in like manner. As the pressure exerted on any structure by a wave depends upon its velocity, the longer the waves the greater the damage that may be created.

The highest waves probably are never greater than 60 feet; and under most circumstances and in most places they are considerably less than this. Such waves usually occur in mid-ocean or away from harbors, so that they need not be considered in their destructive effects on harbor works.

The intensity of the stroke of waves is due in a great measure to the non-elasticity of the water.

The effect of wave action on beaches is peculiar in that offshore storms cause the beaches to heap up, while onshore gales wear them down. Just what the reason for this is has not been fully determined, but it is known that the composition of a beach affects this action to a considerable extent.

Tides. *Theory.* The word tide is used to denote the regular rise and fall of the water in the oceans which occurs twice a day. The theory of the subject is somewhat involved, but it may be briefly stated that the tides are the result of the force of gravity of both the

sun and the moon acting upon the ocean. As the moon is much nearer the earth than is the sun, its effect is correspondingly greater, in fact about 2.3 times that of the sun, and, as a consequence, as ordinarily understood, the moon is generally credited with being the one and only controlling factor with respect to tides.

The effect of this gravitational force, or "pull" of the moon upon the water in the oceans is to raise or pile it up in a plane joining the centers of the moon and the earth, and as the former revolves around the latter, this elevation of the water will follow it. On the side of the earth away from the moon, the same condition of high tide will exist, but at positions midway between these two there will be a stage of low tide.

When the sun and the moon are on the same side of the earth, or on exactly opposite sides, and the same straight line joins their centers with the center of the earth, the pulling force of the sun will be exerted in unison with that of the moon and we shall have the highest possible tides. As the sun and moon lose this relative position, however, the force of gravitation from each will tend to counteract more and more that of the other, so that when the moon is 90 degrees away from the sun their opposing effect on the tides will be a maximum.

Spring tides, therefore, are those at which the water rises highest, due to the combined pull of moon and sun; while *neap* tides are those high tides when the sun's pull is opposing the pull from the moon. As high water of spring tides rises higher than that of neaps, so low water of springs generally falls lower.

Range. The *range* of a tide is the vertical rise between low and high water. *Half tide* is the mean distance between high and low water of springs and neaps.

The difference of rise between spring and neap tides varies in proportion to the range of the tides. As an average, it may be taken that the rise of neap tides above low water of spring tides is 75 per cent of that of a spring tide.

When the sun is vertically over the equator, and their paths coincide, the tide-producing effect is at the maximum. At these periods, which occur in March and September, the new moon and full moon also have most influence. In June and December, when the sun is farthest from the equator, the rise is at the minimum.

New-moon tides are about $\frac{1}{70}$ of the total rise higher than full-moon tides.

It must not be understood that the range of the tides is the same at all places; quite the contrary, for local natural physical conditions will greatly influence the rise and fall of the water. Thus the normal rise of the tides in the ocean under the equator where it would be greatest, has been estimated as being between 2 and 3 feet, while the convergence of the shores of the Bay of Fundy causes a range of 60 feet at the head of the bay, though at the mouth it is only 8 feet.

It is clear from the above that an increase in the range of a tide will increase the volume of construction work in connection with harbor improvements of a protective character. As an example, on a certain type of breakwater in a tideless sea, a rise of 7 feet was estimated to increase the volume of rubble 66 per cent; a range of 15 feet increased the quantity 250 per cent, and a range of 25 feet increased it 400 per cent.

Currents. The principal causes of currents are the heat of the sun's rays, and the rotation of the earth on its axis, if we exclude those which are due to the tides. The former are represented, for example, by the Gulf Stream which flows in a northeasterly direction along the east coast of the United States, and, in the neighborhood of Newfoundland, swings across the Atlantic Ocean toward the British Isles; or by the Japan current in the north Pacific Ocean. The currents which most affect harbor works, however, are not these, but the ones due to the tides. These latter never exist unless the tides are interfered with or obstructed in their natural rise and fall. Thus, when the shores converge and the depth decreases, currents of considerable magnitude may be established; or, when headlands interfere, a pounding action may result, which produces a difference in the level of the water surface on the two sides and this in turn creates a current.

This action, together with its effect on shipping, is well demonstrated at the narrow western end of Long Island Sound, called Hell Gate, adjacent to and practically a part of the harbor of New York City. The waters of the Sound are piled up by the tide at this point, and due to the restricted outlet, are unable to flow freely to the south and west into part of New York harbor, called East River. On the

other hand, the waters from the Upper New York Bay and East River have not reached the same elevation as those in the Sound due to a "lag" in the tide, and as a consequence of this difference in elevation a strong and dangerous current exists at this point.

Such facts as these must be kept in mind in designing harbor works, for when they can be utilized they may become the most effective agents in maintaining the required depth of channel, while if they are ignored, they may render the improvements absolutely useless. As an example of what may arise, due to the restriction of the tidal current, we may refer to an improvement on the west coast of Holland. It had been observed at this place that the current due to the tide had a velocity of from 2 to $2\frac{1}{2}$ knots per hour, whereas, when a long pier about 1 mile in length had been extended out from the shore, the velocity in front of the pier head was increased to between 3 and 4 knots per hour, producing a considerable scour at the foot of the outer piles, and making it at least inconvenient if not dangerous for a ship to land.

BREAKWATERS

Classification. Breakwaters may be divided into two classes as follows: (1) vertical type; and (2) mound type. The former, Figs. 27 to 30, including therein those where the face is slightly

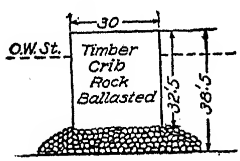


Fig. 27. Section of Breakwater, Harbor of Chicago, Illinois

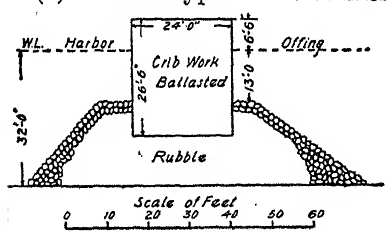


Fig. 28. Section of Breakwater, Harbor of Milwaukee, Wisconsin

inclined, may be constructed of: (1) timber cribwork filled with stones; (2) outer walls, of masonry or concrete, either dry or in cement, with the interior composed of dry rubble; (3) concrete blocks, laid in horizontal courses throughout or in connection with an interior and base of cyclopean character; (4) mass concrete; (5) the same as (4), but with a base formed of concrete in bags; (6) concrete blocks laid in sloping courses; or (7) any of the preceding, with heavy blocks on the exposed face to break the force of the waves.

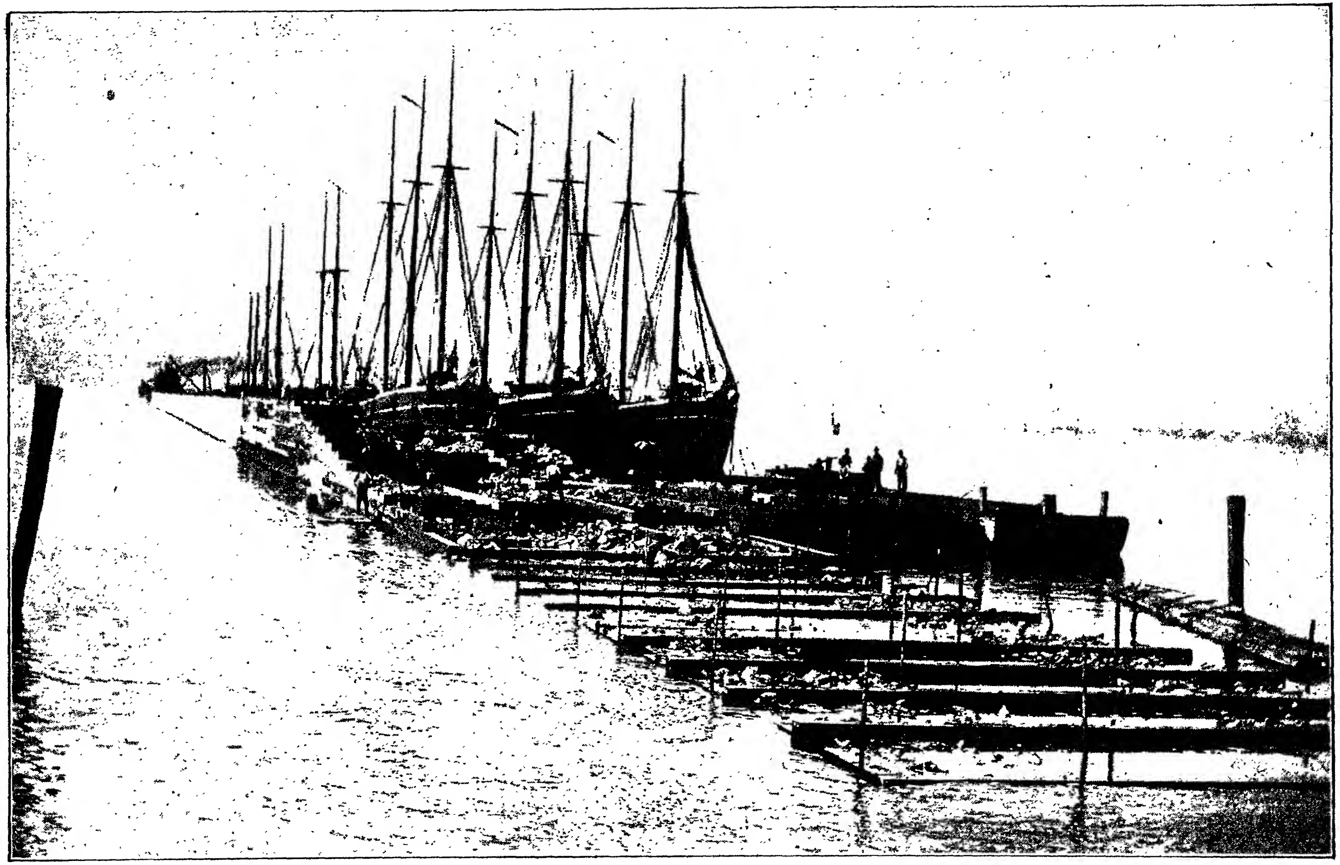


Fig. 29. Improvement of Harbor of Buffalo, New York

Breakwater under construction, showing timber cribwork extension at south end. The substructure cribs loaded with stone, are sunk in place, and the erection of the superstructure is shown under way.

In the mound type, Figs. 31 to 35, they are formed of: (1) masses f rubble stone of all sizes deposited loosely and allowed to take their atural slope; or (2) mounds of concrete blocks deposited loosely, nd brought to high-water level.

VERTICAL TYPES

Peculiar Features. An advantage of the vertical-faced type f breakwater is that it requires less material per unit of length,

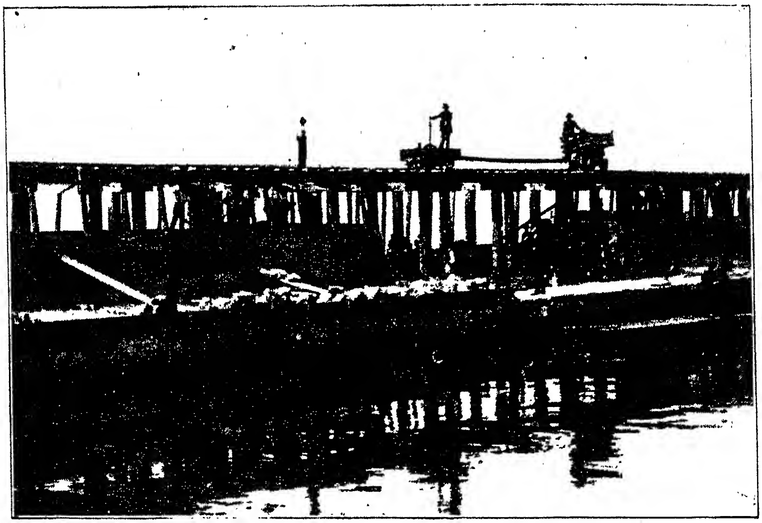


Fig. 30. Improvement of Harbor of Buffalo, New York
Construction of breakwater, showing molds for forming parapet and banquette of concrete superstructure.

and therefore is the one usually selected where material is scarce. Other factors, however, may control in the design, such as depth of water, or nature of foundation; and these must of course always be most seriously considered.

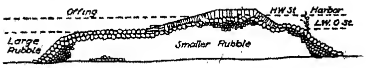


Fig. 31. Section of Breakwater, Harbor of Plymouth, England

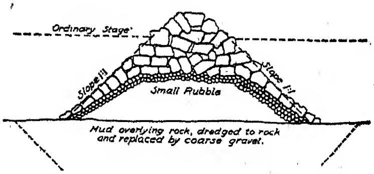


Fig. 32. Section of Breakwater, Harbor of Buffalo, New York

Against a breakwater of the vertical type, waves tend to rise higher than where the other form is employed, so that it may be

stated as a general principle that they are elevated to double the height of their crests above normal water surface. At the same time

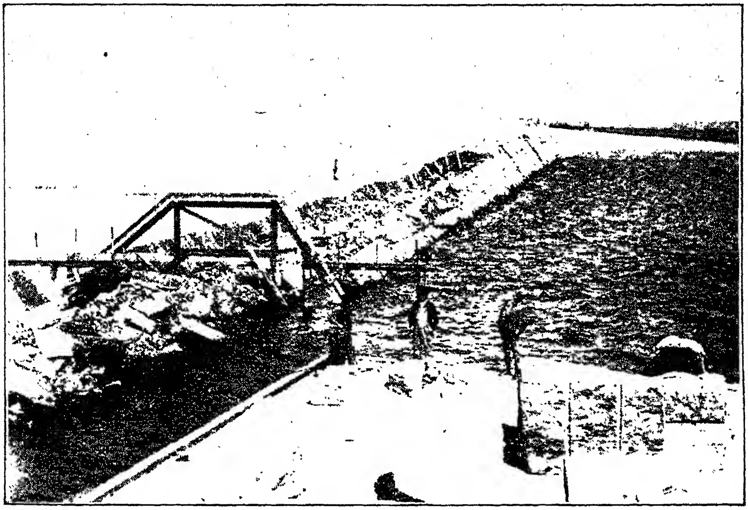


Fig. 33. Improvement of Harbor of Buffalo, New York
Showing method of constructing breakwater.

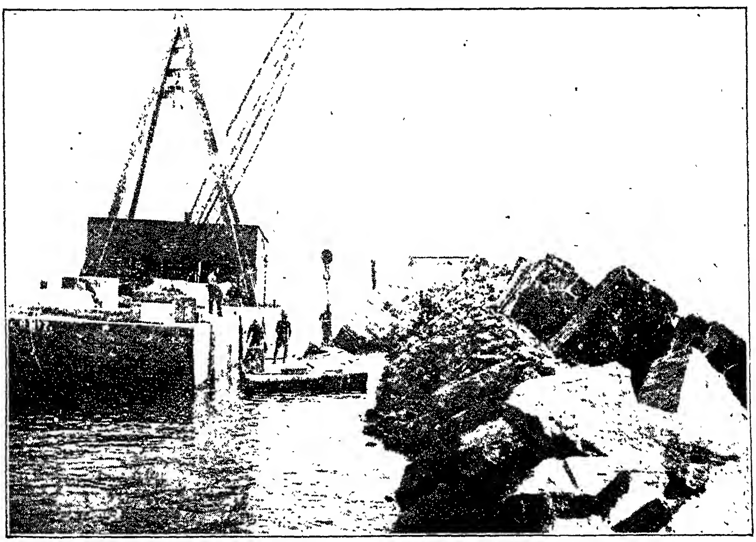


Fig. 34. Breakwater under Construction, Harbor of Buffalo, New York
Showing size of and method of placing facing stone.

that this occurs, a wave is sent downward along the wall, which strikes the bottom and scours out the material of the foundation.

The consequence of this action is that vertical-faced breakwaters are subjected to tremendous forces during the heaviest storms, and

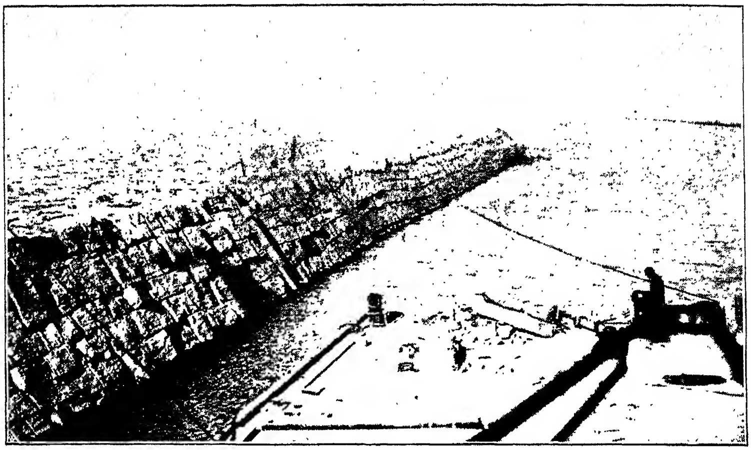


Fig. 35. Breakwater, Harbor of Buffalo, New York, with Facing and Top Stones Complete

damage frequently results, with which the mound type of breakwater does not meet.

Timber Cribs Filled with Stone. These are cheaper than breakwaters of stone, and are more rapidly constructed. Such are the breakwaters at Boulogne, Fig. 36, and at Calais, France, and at Dover, England. Frequently these are constructed as temporary affairs until more permanent ones may be built of stone, and are

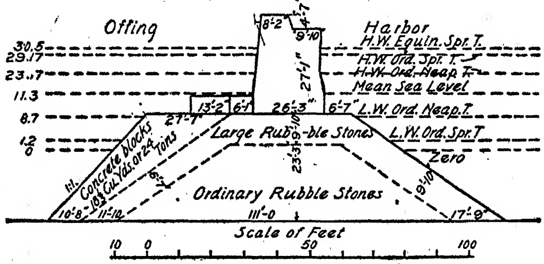


Fig. 36. Section of Breakwater, Harbor of Boulogne, France

usually quite satisfactory. It is imperative that the timber work receive careful attention, for that which is above low water is subjected to decay, while that below may be subjected to the attack of

wood-boring animals (notably the *teredo navalis*), as is well known to be the case on the Pacific Coast.

This type of breakwater is best suited to shallow water, or where the wave action is less intense than it is in exposed positions. They are frequently formed by piles driven into the bed of the harbor, with stone filled in among them. The purpose of the timber is always to hold the stone in place, while the stone itself supports and strengthens the crib.

Type with Masonry Side Walls and Rubble Interior. As in the previous class, this is best suited to harbors where the exposure is slight. The method of construction is to build the exterior walls first, and fill in between them with loose stone which lends bulk and stability to the structure; but, as this leaves the outside shell unsupported for a considerable length during construction, it permits of the possibility of the breakwater being destroyed before completion, should a heavy storm arise. Furthermore, masses of water falling from above on the unprotected rubble interior, fill it with water and exert an excessive outward lateral pressure which may cause it to fail in its purpose.

Concrete-Block Type. This method possesses several advantages over the use of loose stone or block stone put in place, because each block is better able to withstand the wave action than an equal amount of small material.

In this type it is very necessary that the foundation be leveled off and properly prepared, as the first course of concrete blocks must be laid horizontally, and have close joints. The blocks are placed in position by the aid of divers, those blocks below the water being laid without mortar. Owing to this fact, the blocks are less likely to bear against each other uniformly, and hence they tend to break. For the same reason (i.e., lack of cement in the joints), no tendency toward settlement should be permitted, as this would allow the blocks to slide or move under the force of the waves.

Cement or mortar is generally used between the joints when the masonry has risen above the low-water mark; and the topmost course, or capping, is usually formed of mass concrete deposited in place.

Sometimes, in this type, the foundation is formed of mass concrete which is brought up to the level of low water; and then, on top of this, are placed the concrete blocks laid in mortar.

Mass-Concrete Type. This system consists in depositing wet concrete in temporary timber frames below or above water level, and is employed on both large and small work alike. The expense and time required to level off the foundation in the previous system, are obviated in this. It has the disadvantage, however, of being essentially a fine-weather method of construction, as both the frames and the wet, or "green", concrete will not withstand storms.

To prevent the loss of cement during deposition, the concrete is frequently put in place by loading it into skips with a flap bottom, lowering this under water to within a few feet of the bottom, and then opening the skip so that the concrete is discharged. As little disturbance as possible of the concrete and of the water in which it is to rest, is the most desirable condition. This system is generally preferred, though sometimes the concrete is deposited by using a chute. To keep the water out, the chute should be maintained full of concrete all the time.

For purposes of economy, large pieces of stone may be incorporated with the concrete; but the amount should never exceed about $\frac{1}{6}$ of the entire mass, and the stones should always be separated from each other by at least 9 inches or a foot of concrete. The concrete should have the proportions of about 1 part of cement to 3 or 4 parts of stone.

Concrete-Bag Type. This system is sometimes employed for underwater work, and consists simply in placing the concrete in canvas bags, so as to keep it and the cement in place. By this means an even foundation may be secured, even with an irregular bottom.

Such construction is frequently employed as an apron to prevent the scour of foundations, the bags being deposited by means of skips with bottoms opening outward and of a capacity of from 5 tons to 20 tons. In the construction of the Aberdeen breakwaters, Scotland, bags of concrete have been used weighing as much as 100 tons. The richness of the material used is generally about 1 part cement to 4 or 5 parts sand and broken stone.

The material should be deposited in place as soon as possible after mixing, so as to prevent it from acquiring a set, and, when so deposited, it should be lowered as close as possible to the place where it is to rest, before being released. Dropping the bags through considerable heights is to be avoided as much as possible.

This system is best adapted to localities but little exposed, or where the foundations are deep, as the bag work is not well suited to resist the wave action. It may be so employed, however, and even brought up to an elevation above low water, if the exposed face is protected by a deposit of loose blocks of stone.

Type with Concrete Blocks Laid on Inclined Courses. This system consists in laying blocks of concrete with the joints at an angle to the vertical, and may be used either where the breakwater is to have vertical faces or where there is to be a loose-stone base. The foundation must first be leveled off in this case, which is accomplished by means of mass concrete in bags, or by the use of broken stone.

Blocks weighing from 20 tons to 120 tons each have been employed in this system. The amount of inclination of the blocks will vary, but a slope of between 10 and 30 degrees from the vertical usually proves satisfactory. Frequently the blocks are arranged with a sort of tongue and groove which fit into each other to secure greater stability. The object to be attained in this method is the closeness of joints and the prevention of the slope from flattening.

As a system it compares very favorably with others of this class, since there is little risk during construction, little injury from settlement, and possible rapidity of erection.

MOUND TYPES

Loose Stone. As a mound-type breakwater requires an excess of material per unit of length, it is absolutely necessary that the materials of construction be abundant.

The effect of the tides on such structures being very considerable, it becomes more and more necessary, as the range increases, to enlarge the cross-sectional vertical area of the breakwater. The waves sweeping over such structures tend to change the slope of the rubble stone more and more, until the material has at last, perhaps after years, come to rest in accordance with the forces of nature. This slope will depend not only upon the height of the waves, the depth of the water, etc., but upon the material itself and the height to which it is carried above low water.

It is found that the waves on the harbor side affect the slope of this material to a depth of 10 or 15 feet below low tide, and the tendency is to reduce it almost to the horizontal or to as small a slope

as 1:72. At Plymouth, England, the breakwater on its outer face has the following slopes: below low tide, a slope varying from $1:3\frac{1}{4}$ to 1:5 to a depth of about 10 feet; and from there to the bottom, where the wave action is ineffective, it assumes that of the angle of repose of the material, which is about $1:1\frac{1}{2}$. Above low tide, the slope is almost horizontal, extending right up to the high-water mark; on the harbor side, the slope is approximately that of the angle of repose.

Where a vertical wall is placed on top of such a breakwater—for example, for the purpose of forming a quay—the tendency for the action of the water to disturb the material in the mound only increases, which tendency does not exist when the wall is absent. This is caused by the downward action of the waves and the force of recoil.

The best practice would indicate that a rubble mound surmounted by a superstructure should either be carried well above high water, and be maintained at that level, so as to make the waves break and expend themselves on it; or that it should be kept at such a depth below low water as not seriously to affect the character of the waves, or to be exposed to their disturbing action.

In rubble mound breakwaters, it is probably better to mix the material as to size than to have it uniform, as in the latter case the interstices left are too large and permit of wave action therein. The largest stone should, however, be placed on the outer slope.

Concrete Blocks Thrown Together Irregularly. Not many of this type have been built, as the expense is too great; but they are often combined with rubble, to form the upper part or the outer face of the breakwater. They are found to some extent in the harbors bordering the Mediterranean Sea, particularly in Italy.

Where the blocks are large, as in this type, they take a steeper slope than rubble; and, having more voids, they resist and check the wave action better.

SIGNALS

The function of providing lights and buoys for a harbor generally devolves upon the local authorities, though in some cases it may resolve itself into a national undertaking.

The necessity for such lights and buoys to mark a channel or indicate the shoals on a coast, is readily appreciated when it is realized that vessels now proceed into port under full steam whether it

be day or night, and consequently a channel must be made as easy to navigate in the dark as in the daylight.

Code of Regulations. Unfortunately no uniform set of laws exists for the world nor even for the same country in regard to the lighting and buoying of harbors. While this is of little consequence where a vessel is being guided by a local pilot, under other circumstances it is sometimes a hardship, particularly to smaller-sized vessels visiting the port. In order to obviate the difficulty somewhat, the following rules are in force wherever the British Admiralty, Board of Trade, and Lighthouse Board have jurisdiction.

In framing these regulations, the shape of the buoys was relied upon more than the color, as distinguishing the different parts of the channel. There is a well-known rule that the right bank of a river is the one upon the right hand when going *down* the stream from source to mouth. It is unfortunate, being likely to lead to confusion, that this should have been abandoned in favor of the stipulation that the right-hand side is that on the *right* when going *up* the channel.

REGULATIONS FOR BUOYING CHANNELS

The term *starboard hand* shall denote the side which would be on the right hand of the mariner either going with the main stream of flood or entering a harbor, river, or estuary from seaward; the term *port hand* shall denote the left hand of the mariner under the same circumstances.

Buoys showing the pointed top of a cone above water shall be called *conical*, and shall always be starboard hand buoys, as above defined.

Buoys showing a flat top above water shall be called *can*, and shall always be port hand buoys, as above defined.

Buoys showing a domed top above water shall be called *spherical*, and shall mark the ends of middle grounds.

Buoys having a tall, central structure on a broad base shall be called *pillar* buoys, and, like other special buoys, such as bell buoys, gas buoys, automatic sounding buoys, etc., shall be placed to mark special positions either on the coast or in the approaches to harbors, etc.

Buoys showing only a mast above water shall be called *spar* buoys.

Starboard hand buoys shall always be painted in one color only.

Port hand buoys shall always be painted of another characteristic color, either single or parti-color.

Spherical buoys at the ends of middle grounds shall always be distinguished by horizontal stripes of white color.

Surmounting beacons, such as staff and globe, etc., shall always be painted of one dark color.

Staff and globe shall only be used on starboard hand buoys; staff and cage on port hand; diamonds at the outer ends of middle grounds, and triangles at the inner ends.

Buoys on the same side of a channel, estuary, or tideway may be distinguished from each other by names, numbers, or letters, and, where necessary, by a staff surmounted with the appropriate beacon.

Buoys intended for moorings, etc., may be of shape or color according to the discretion of the authority within whose jurisdiction they are laid; but for marking submarine telegraph cables, the color shall be green, with the word *telegraph* painted thereon in white letters.

Wreck buoys in the open sea, or in the approaches to a harbor or estuary, shall be colored green, with the word *wreck* painted in white letters on them.

When possible, the buoys shall be laid near to the side of the wreck, next to mid-channel.

When a wreck-marking vessel is used, she shall, if possible, have her top sides colored green, with the word *wreck* in white letters thereon, and shall exhibit: *By day*—three balls on a yard 20 feet above the sea, two placed vertically at one end and one at the other, the single ball being on one side nearest to the wreck; *by night*—three white fixed lights similarly arranged, but not riding light.

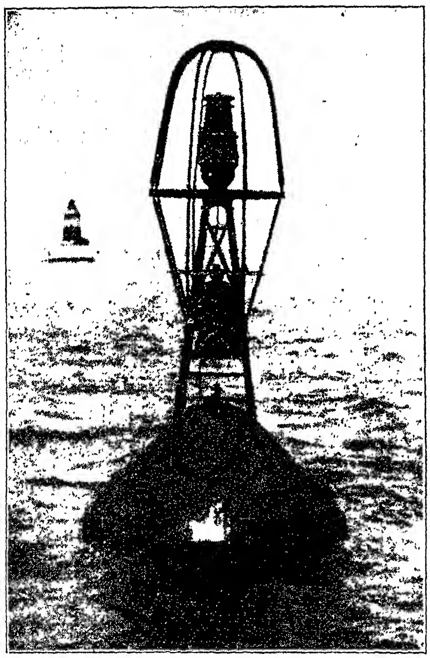


Fig. 37. Gas-Lighted Bell Buoy

In narrow waters, or in rivers, harbors, etc., under the jurisdiction of local authorities, the same rules may be adopted, or, at discretion, varied as follows:

When a wreck-marking vessel is used, she shall carry a cross-yard on a mast, with two balls by day placed horizontally not less than 6 feet nor more than 12 feet apart, and two lights by night similarly placed. When a barge or open boat only is used, a flag or ball may be shown in the daytime.

The position in which the marking vessel is placed with reference to the wreck, shall be at the discretion of the local authority having jurisdiction.

Buoys. There are various kinds of buoys, the following being the better-known—*can, cone, nun, spherical,* and *pillar;* and they are so designated, depending upon their shape.

The *can* buoy has the cone immersed; and the base with a flat top is the part that is seen above the water.

The *cone* buoy is practically a metal cone, which floats with the base in the water, and the cone projecting above.

The *nun* buoy is pointed at both ends, with the largest diameter at water level, and because of its lightness and simplicity, is particularly useful in shoal water.

The *spherical* buoy has a sphere showing above water.

The *pillar* buoy is practically the same as a can buoy with a pillar resting on top of it.

Bell buoys, Fig. 37, are used to mark dangerous parts in the channel, or shoals, and have the advantage over the others of being as useful in foggy as in clear weather.

These buoys are now universally made of steel, and most of them exposed to heavy seas are divided into two compartments so that in case of injury to the one, the other will maintain it in its position.

The purpose of these various-shaped buoys is to expose as much of the body of the buoy as possible, so that it will be easily distinguished; and it is a question which shape accomplishes this best. Long, narrow buoys have a tendency to lie over in the flowing tide, while flat-bottomed ones seem less inclined to do so.

All buoys should be cleaned and painted at least once a year, and oftener, where conditions require it. Tar withstands the salt water better than paint, and a coat of neat cement is still better, as it absolutely protects the metal of the buoy.

Buoys are maintained in position, by chains attached to anchors or sinkers. The chain is usually from 2 to 3 times the depth at high water, the reason for this being that it is then less subject to sudden jerks during bad weather. The sinkers used are generally of cast iron, and weigh from 1400 to 1600 pounds, while the anchors are of the "mushroom" type. A buoy, when properly moored, seldom breaks adrift. The greatest danger arises from ice, when large drifts are sent out of the rivers at the breaking-up of a long and hard frost.

Lights. For lighting the channel leading to New York harbor between Sandy Hook and Coney Island, a system of electric buoys has been adopted. The buoys, made of wood, 50 feet long and 15 inches in diameter (to be replaced by buoys of riveted steel), are shackled to a cast-iron sinker weighing 2 tons. A cable from the shore is led up to an electric light on top of each of the buoys, the power being supplied from one station at Sandy Hook.

Efficiency of Illuminants. From some 6000 observations made by the English Lighthouse Board to determine the relative efficiency

of the various illuminants—oil, gas, and electricity—the following conclusions have been drawn: (1) that the electric light was the most powerful under all conditions; (2) that the quadriform gas apparatus and triform oil apparatus were of about the same power when seen through revolving lenses, the gas being a little better than the oil; (3) that through fixed lenses, the superiority of the gas lights was unquestionable, the large size of their flame and their nearness together giving the beam a more compact appearance; (4) that the Douglas gas burner was more efficient than the Wigham burner;

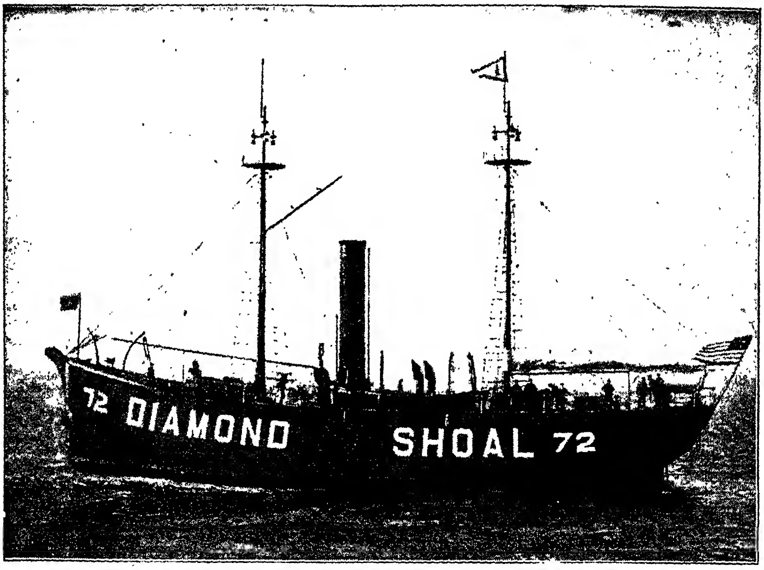


Fig. 38. Diamond Shoal Lightship

Moored in about 180 feet of water, about 14½ miles off Cape Hatteras Light Station. It has a 12-inch steam chime whistle, which blows a 5-second blast, with silent intervals of 55 seconds.

(5) that for the ordinary necessities of lighthouse illumination, mineral oil was the most suitable and economical illuminant; and (6) that for salient headlands and places where a very powerful light was required, electricity offered the greatest advantages.

Usage. Where harbors are approached by a channel, it is usually customary to mark such by buoys, and by lights at night. These are separated at a distance apart of from 0.1 of a mile to 1½ miles, depending upon the circumstances, and are so placed as to indicate the channel, shoals, bends, obstructions, etc.

Lights are divided into sizes depending upon the height and diameter of the light. The first three sizes are for the sea or coast, while the fourth size is the largest used for a harbor.

To avoid mistakes, a red tinge may be given the light by using a colored chimney or a sheet of colored glass; but it has been proven that a white light can be seen much farther than a red one, and a red light farther than a green.

The height in feet at which a light should be placed above sea level to be seen a given number of miles away, is determined by multiplying the square of the distance in miles by $\frac{4}{7}$. Thus, if it is desired that a light be seen 15 miles away, it must be placed at a height equal to $15^2 \times \frac{4}{7} = 225 \times \frac{4}{7} =$ 128.6 feet.

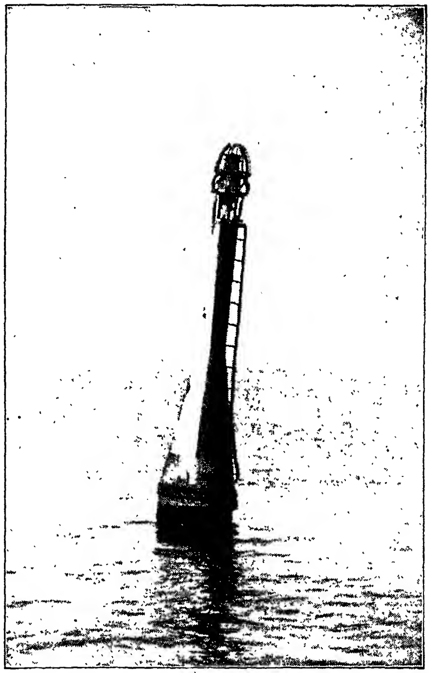


Fig. 39. Spar-Shaped Gas Buoy

Originally, lights used as signals were made by burning wood on the exposed or dangerous headlands; but today, either oil, gas, or electricity is used in some sort of lantern which increases the power of the ray enormously. Thus, the burner invented by Sir James Douglas is able to increase the intensity 10 times.

Various kinds of oil have been used in such lamps; but mineral oil seems to satisfy the needs best. It gives a more brilliant light than vegetable oil, does not congeal at low temperature, and requires less trimming of the wicks.

Classification. In harbors the lights used are: *beacon lights,* to mark prominent headlands; *floating lights,* to mark the channel; and *leading lights,* to mark the entrance to a river, or the course in a channel.

Floating lights (that is, *lightships*), in English waters, are always painted, and have a ball at the masthead. Such vessels are

usually of small tonnage, and are anchored by means of mushroom anchors. During fog, horns, bells, or gongs are sounded to indicate the position. An American lightship is shown in Fig. 38.

Gas buoys, Figs. 37 and 39, are frequently used by employing

Fig. 40. Cape Hatteras Light Station, North Carolina

a compressed oil gas stored in a reservoir in the buoy, and which, burning day and night, may last for weeks without replenishing.

Lighthouses. Lighthouses are usually towers built of iron or masonry, well above the height to which waves may reach, and in such a manner that the force of the waves may not destroy the struc-

ture. The waves that assail them are always of less intensity than those found in the open sea, as the location of a light is always

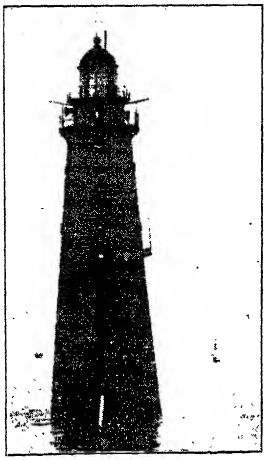


Fig. 41. Minot's Ledge Lighthouse, at Entrance to Boston Harbor

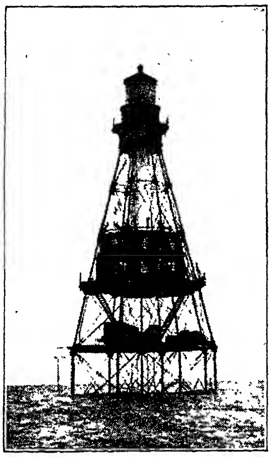


Fig. 42. American Shoal Light Station, Florida

placed on some rock-bound coast or shoal, near shore. It is necessary in their construction, however, to know the height of the waves in

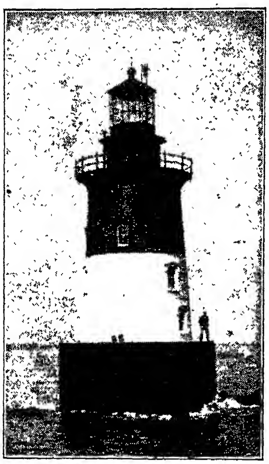


Fig. 43. Romer Shoal Lighthouse, New York

Fig. 44. Rebecca Shoal Lighthouse, Florida

relation to their fetch—the distance along which they are propagated—and also to determine their effect. The rate of increase in the height of waves has been found to be proportional to the square

root of the distance from the windward shore; and the greatest force ever said to be exerted on exposed rocks is estimated to be $3\frac{1}{2}$ tons per foot. The force of winter gales is about 3 times that of summer.

One of the best-known lighthouses in the world is the Eddystone Light, which lies on the Eddystone Rocks about 14 miles from Plymouth, England. It was one of the first lighthouses of the building of which we have record, and has been reconstructed many times. In 1698 it had a height of 80 feet; but owing to damage, it had, in the following year, an extra shell of 4 feet built around it. In 1700 the height was increased to 120 feet, but during the year 1703 the tower was destroyed. In 1706 it was reconstructed, then having the form of a frustum of a cone, and being 92 feet high. It was principally of timber construction, the lower part being of oak. Above this a stone structure rose. It rested upon a stepped level base, was circular in plan, depended for fixedness upon its weight, and had a uniform exterior. In 1756 the light was rebuilt by Smeaton, being completed in 1795. It was entirely of masonry, the first of its kind, with dovetailed joints, each stone weighing about 1 ton. In 1882 a new tower was erected. Typical lighthouses of this and other countries are shown in Figs. 40 to 47.

Fig. 45. Hog Island Shoal Lighthouse, Rhode Island

General Rules of Construction. The design of one lighthouse tower will not serve as a model for any other, except under identical conditions. The following rules may be considered fairly general:

(1) The tower should have a low center of gravity, and sufficient mass to prevent its being upset by the waves.

(2) It should be throughout circular in the horizontal plane, and either straight or continuously curved in the vertical plane, so as to present no abrupt change of outline which would check the free ascent of the rising waves, or the free descent of the falling waves, or the free vent of those passing round the tower. All external scarcements in the vertical plane, or polygonal outline in the horizontal plane, are therefore objectionable.

(3) Its height should be determined by the distance at which the light requires to be seen by the sailor. The rule for determining this has already been given.

(4) Where the rock is soft, or consists of ledges which are easily torn up, the tower should spring from the foundation course, at a low angle with the surface of the rock, so as to prevent its being broken up by the reaction of the waves from the building; or, in other words, the tower must have a curved profile. But special care should be taken to sink the foundation courses below the surface of the rock, as the superincumbent weight decreases with the sine of the angle of inclination of the wall. If the rock overhangs, owing to the wearing action of the waves, the tower should, if possible, be built at a distance from the place where this dangerous action is in progress.

(5) Where the rock is hard and of ample area, the tower may be of such a curved form as will best suit the economic arrangement of the materials, so as to avoid an unnecessary thickness of the upper walls.

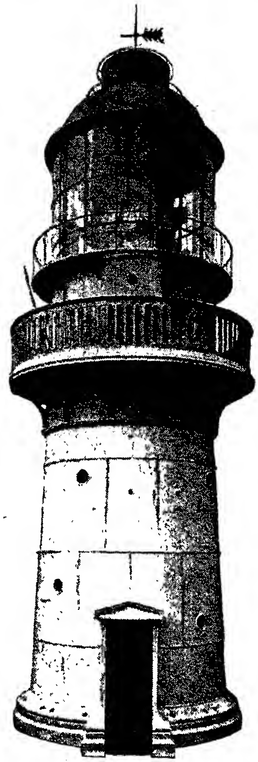


Fig. 46. Lighthouse for Promontory of Lao T'Ieh Shan, China

(6) When the rock is hard, but of small dimensions, the diameter above the base should not be suddenly reduced; and if the rock is hard, but of yet smaller dimensions, a cylindrical form of greater height should be adopted, so as to thicken the walls and to increase the weight and therefore increase the friction, which is directly proportional to the weight of the blocks of masonry. In all cases where the rock is small, the thickness should be decreased by steps or scarcements, *internally* but never *externally*.

(7) The level of the top of the solid part of the tower, and the thickness of the walls above it, should, in different towers having the same exposure, be determined in each case by the level and configuration of the rock and of the bottom of the sea.

(8) The best position of the tower is not necessarily the highest part of the rock. It should, in each case, be selected so as to secure the greatest protection in the direction of the maximum fetch and deepest water near the reef.

(9) The tower, if possible, should not be erected across any chasm which divides the rock, nor in the direction of any gully which projects into it, especially if it be of converging form, which would concentrate the wave action.

(10) No permanent fixture of the tower to the rock is required for increasing the stability of the structure. The foundation course (unless where a curved profile is adopted) becomes, indeed, in the end the most stable of all, because it has the greatest weight above it to keep it in its place.

(11) The stones should, however, be sufficiently connected with each other and fixed to the rock, in order to prevent their being washed away during

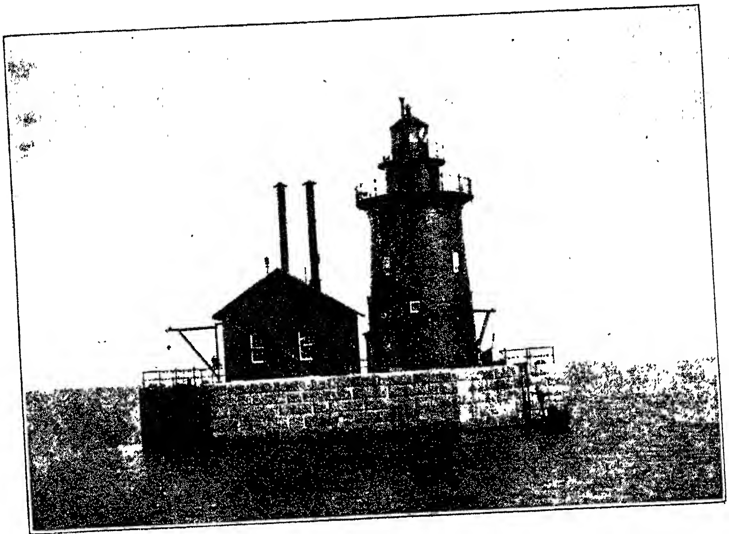


Fig. 47. Bar Point Light Station, Detroit River

the construction of the work, when they have no superincumbent weight to keep them in their beds.

(12) The tower should rest on a truly level base, or on level steps cut into the rock.

(13) The pressure of all the materials within the tower should act vertically, so as not to produce a resolved force acting laterally as an outward thrust.

(14) The tower should be of such height and diameter, with walls of such thickness, as to prevent the masonry being disturbed by the impact of the waves.

(15) The entrance door should be placed on that side of the tower where the length of the fetch is shortest, or where, from the configuration of the reef and the depth of water, the force of the waves is least.

(16) The materials should be of the highest specific gravity that can be readily obtained; and, in some special cases, lead, or dovetailed blocks of cast iron set in cement, might perhaps be employed.

Illumination. The purpose is to distribute the rays equally, either constantly or periodically, over the entire horizon or else over certain angles of the horizon.

Fixed Light. A fixed light is one which is always of the same intensity and constantly in view about the whole horizon. The apparatus for such a light should therefore bend the rays in a vertical

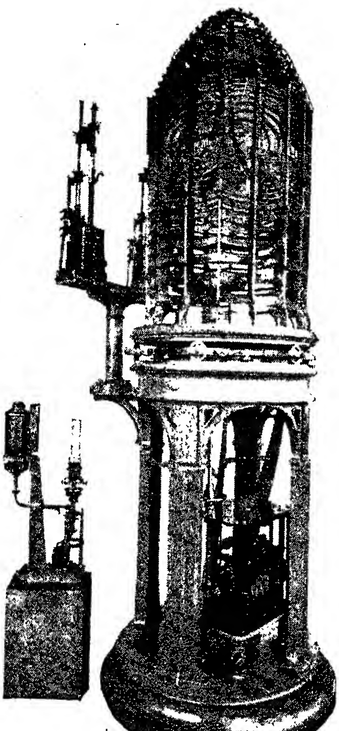


Fig. 48. Light for Lighthouse on Cape Villano, Spain
Showing apparatus giving double flash every 15 seconds.

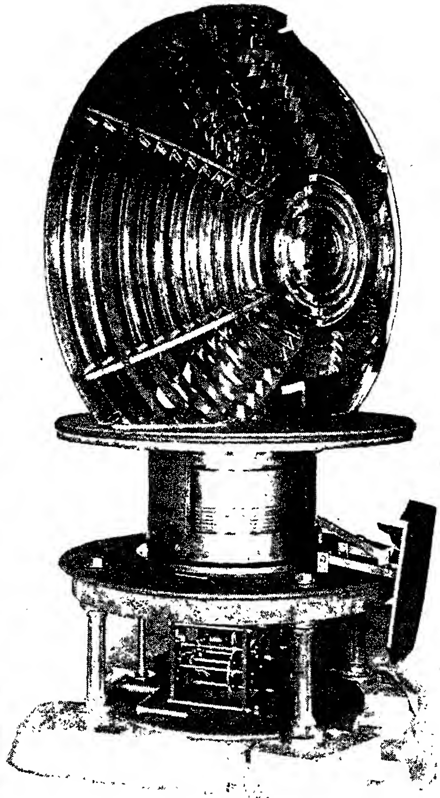


Fig. 49. Third-Order Lightning Light, Toledo (Ohio) Harbor
Fresnel lenticular apparatus, showing clockwork for revolving.

plane, but should not interfere in the horizontal one, if the rays of light are to be sent in the direction desired—i.e., downward and outward over the water.

Revolving Light. A revolving light, on the other hand, illuminates each point of the horizon successively. A revolving light, though supplied by a flame of the same intensity as a fixed light, will

therefore be far stronger than the latter, as it does not lose power by being diffused, but gathers the rays into beams of greater intensity. Metal or glass agents are usually employed in all lighthouses; but by the use of glass alone, it has been found that fully 25 per cent of light may be saved, Figs. 48 and 49.

Beacons. The beacons have done eminent service. They are constructed generally of stone, of rubble, or of iron columns set into rock. The first light of this nature was constructed in 1851, on a sunken reef in Stornoway Bay, Hebrides Islands, Scotland.

These beacon lights are frequently kept burning either by oil or gas, for weeks at a time without being trimmed. Pintsch gas is often employed for them, a buoy or a beacon being charged with gas sufficient to burn night and day for three or four months, the pressure being kept constant by an automatic regulator.

DOCKS

Dry Docks. Docks may be classified as either *dry* docks or *wet* docks. A dry dock, Figs. 50 to 53, is a water-tight chamber, supplied with gates which, when the vessel has assumed its position in the dock, are closed tightly and the water pumped out, thus leaving the hull of the ship exposed so that repairs may be made. With this type, the ship is floated into position; while with some other forms, Figs. 54 and 55, the dock itself is submerged, as a sort of caisson, and when in a proper location beneath the ship to be docked, the water is gradually exhausted, air admitted, and the boat raised.

Dry docks—or *graving docks,* as they are frequently called—vary in length and width, depending upon the size of vessel they are supposed to accommodate. One of them has been built as long as 860 feet, and 90 feet in bottom width. The sides of the dock consist of a series of masonry steps called *altars,* against which timbers are placed to support the vessel in position while the keel rests on blocks.

Such docks are usually constructed of masonry, though they are sometimes built of wood. It is particularly necessary, whatever the construction, that care be exercised to prevent leakage.

Floating Dock. Dry docks have been constructed of steel, and are then usually in the form of caissons which with the water exhausted, may be floated beneath the vessel, to raise the ship; they

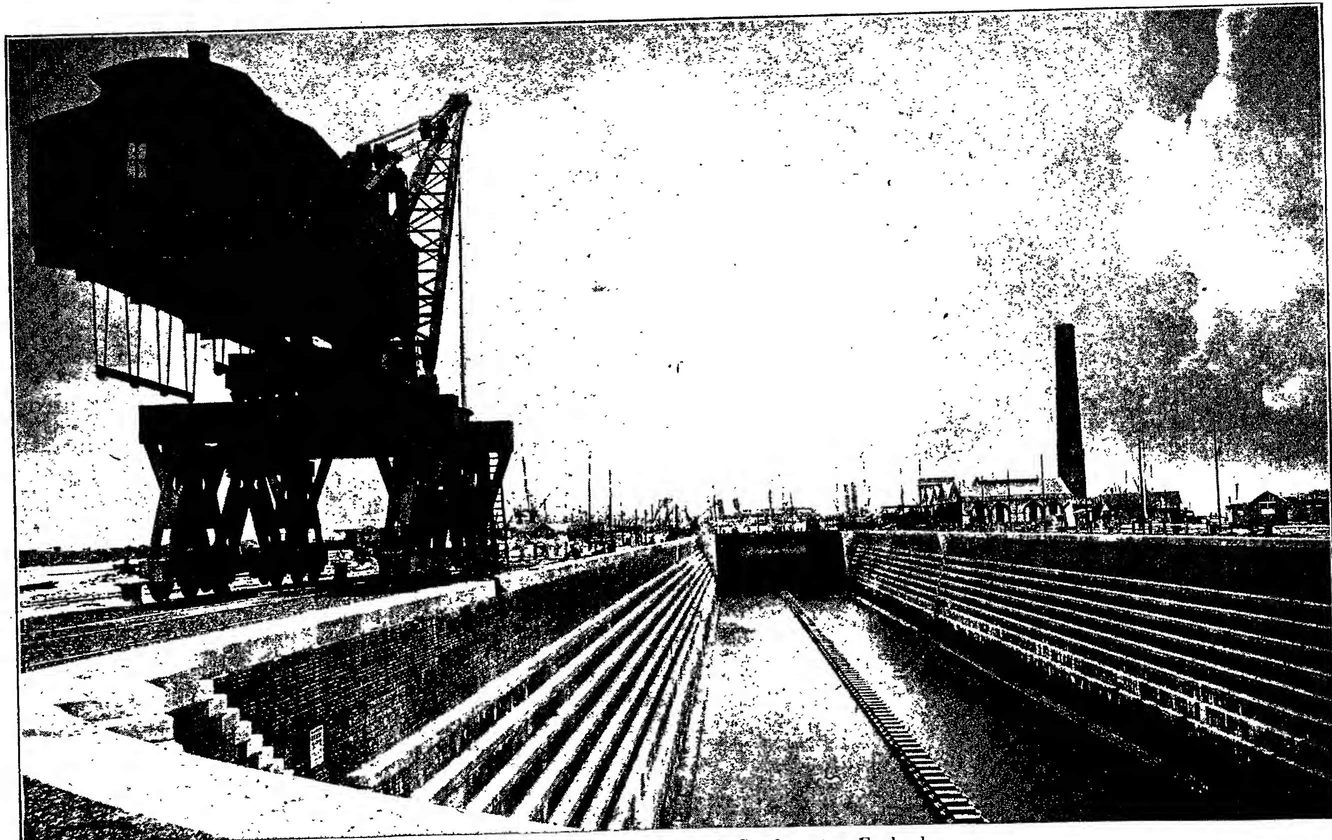


Fig. 59. Prince of Wales Dry Dock, Southampton, England

One of the six large docks owned by the London & Southwestern Railway Company. At the time of its construction, the Prince of Wales Dock (opened about 1896 by His Majesty, King Edward VII, then Prince of Wales) was the largest graving dock in the world. At the same port, a larger dock (begun in 1899) has since been constructed, 860 feet long (clear inside gates), 125 feet wide at cope level, tapering to 90 feet wide at bottom, 43 feet deep, giving a flotation depth over keel blocks of from 29 feet 6 inches, neap tides, to 33 feet, spring tides, at high water.

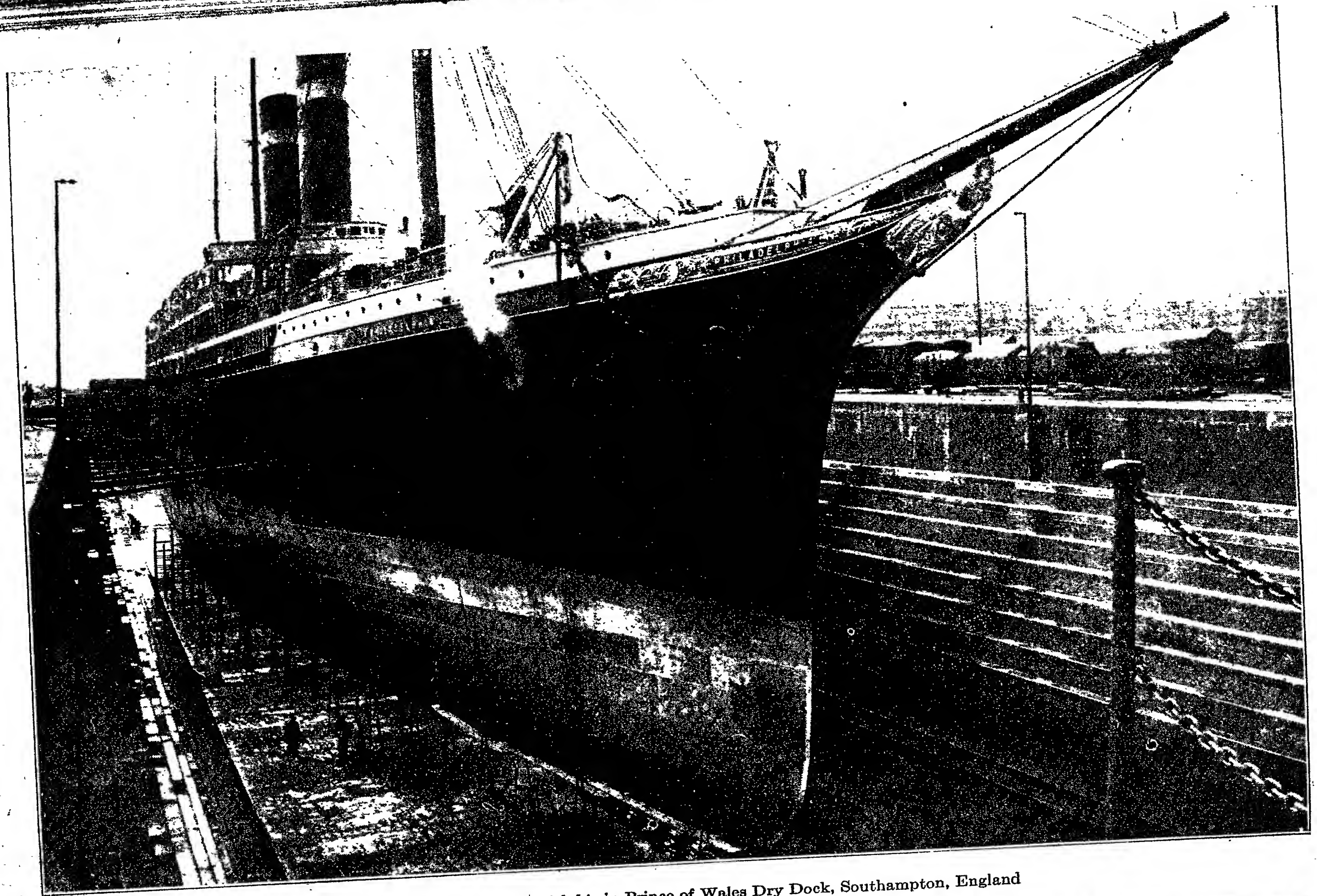


Fig. 51. Steamship *Philadelphia* in Prince of Wales Dry Dock, Southampton, England

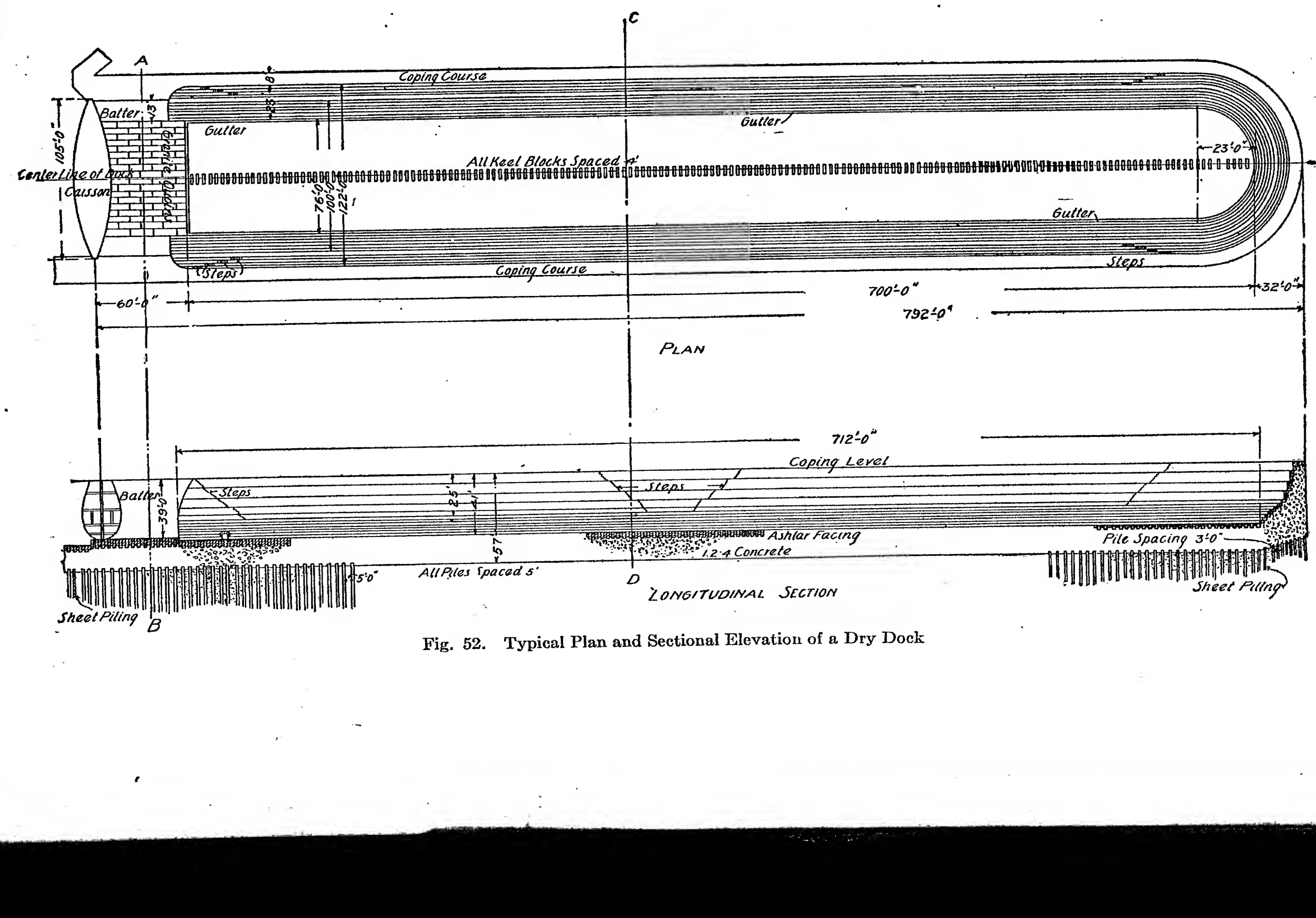


Fig. 52. Typical Plan and Sectional Elevation of a Dry Dock

are then called *floating* dry docks. The particular advantage of this latter form is that it is independent of the tide.

Generally, they are designed for vessels of small register from 2000 to 4000 tons, though the one at Hamburg, Germany, the largest of its kind in the world, has a lifting capacity of 35,000 tons. It was built in 1909, is of steel, 720 feet long, and 108 feet wide.

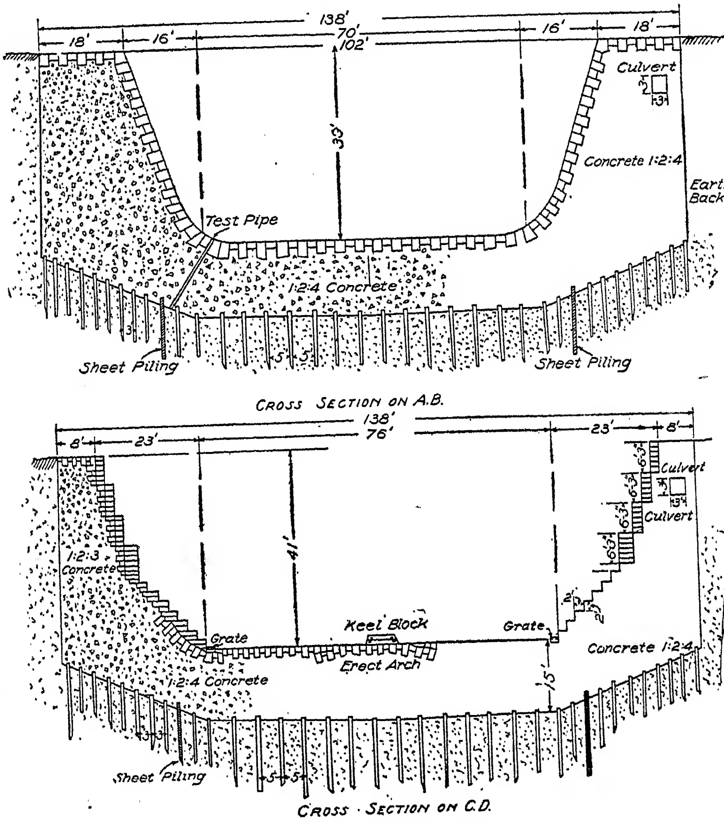


Fig. 53. Transverse Sections of Dry Dock
See plan and elevation on opposite page.

Advantages of Graving Docks. The general advantages connected with graving docks as compared with slips are as follows: (1) Although costly, they are more durable, if well built, than other methods for docking vessels. (2) They require little supervision or management, as they are easily controlled. (3) They are easily worked, whether there be large or small vessels using them.

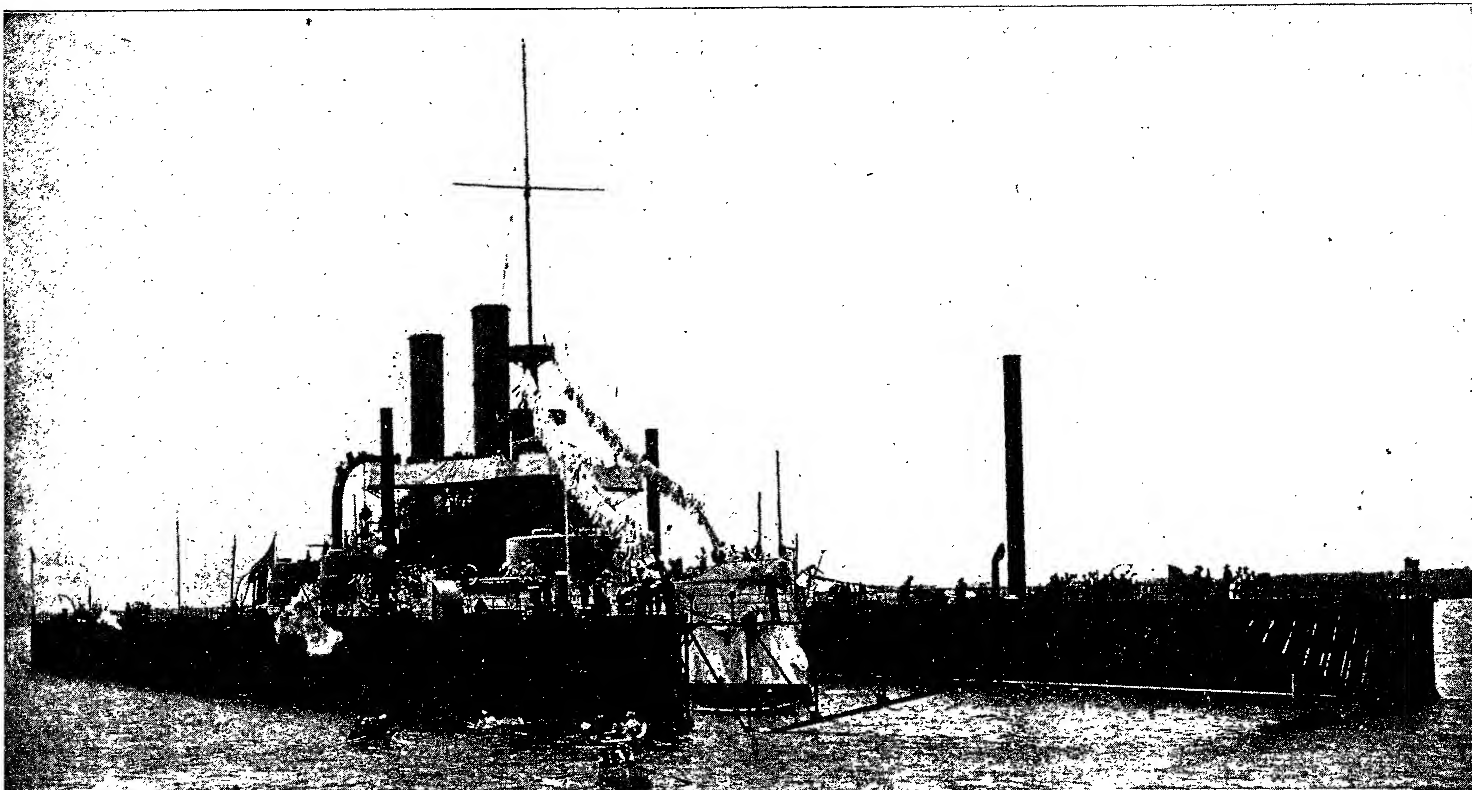


Fig. 54. United States Naval Floating Dry Dock *Dewey* Lifting Battleship *Iowa* out of the Water

The *Dewey* is the largest floating dry dock ever built. Lifting capacity, 18,500 tons with a freeboard of 2 feet. Draft ranges from more than 50 feet to a little over 6 feet. Over 11,000 tons of steel and 2,000,000 rivets used in construction. Cost over $1,125,000. Safely towed in 1906 from Chesapeake Bay on the Atlantic Coast to the naval station at Olongapo in the Philippines, a journey of more than 11,000 miles.

(4) They afford an easier means for filling a vessel with water than any other form of dock—sometimes a necessary undertaking to detect leaks. (5) They tend, with the water that is in them, to scour the forebay or entrance, where double gates are provided. (6) They render it much easier to dock a vessel where the current is rapid or strong. (7) They need not interfere with the set of the currents. (8) They cause less of a strain on the ship.

Slips. Slips consist of a cradle or carriage which is worked on an inclined railway, at various inclinations, and extends some distance above high water. The cradle is let down into the water beneath the ship, which is put in place, and the cradle is then drawn up till it catches the ship, when both are hauled out of the water.

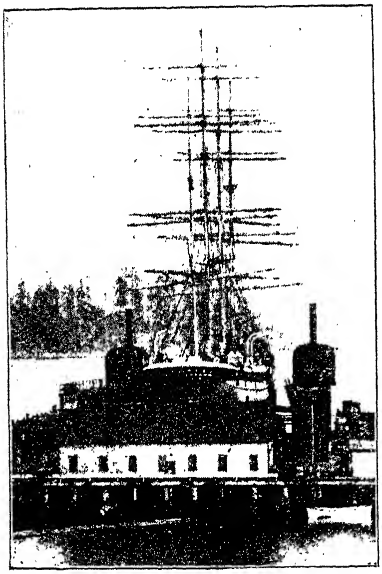


Fig. 55. Type of Floating Dry Dock Used on Puget Sound

When the caissons are air-filled, the dock rises, lifting the vessel out of the water.

Wet Docks. Wet docks are great enclosures or basins provided with a number of piers, quays, or wharves—at which vessels may be berthed for loading and unloading—and furnished with gates which are kept open during high tide, and closed during the ebb, so that the water may be maintained at a constant depth in the basin. That they may be used at all tide periods, locks are often supplied at the entrance so that at low water the boats may be lifted up into the dock. They are employed where the range of the tide is considerable.

Tidal Wet Docks. Wet docks may be divided into a subclass called *tidal wet docks* to designate those which are open to the rise and fall of the tide. They are used where the tidal range is small, and where in consequence, no basin or enclosure with gates is necessary. Fortunately for this country, the variation of the tides is so limited along its sea coast, that expensive wet docks, on the type of those constructed abroad, are not necessary.

In New York harbor, for example, the docks or piers consist simply of rectangular structures, at right angles to the shore line, built on piles as a foundation, running into the water, with a roadway or floor the entire width and length of the structure, level with the street, and over all a wooden or corrugated-iron building furnishing the roof and walls for protection to cargoes that are being loaded or unloaded. Frequently, part or all of the side walls are made of movable iron curtains which may be raised or lowered so that any gangway to the ship alongside the dock may be easily accessible. Here at all conditions of the tide, the largest vessels afloat may come up to the docks under their own steam and, with the assistance of numerous small tugs, be nosed into their restricted berths alongside, where the loading or unloading begins within 15 minutes after their arrival. This obviously gives New York a notable advantage over other ports where wet docks are necessary, with expensive masonry walls to enclose the basin, costly gates and perhaps locks, and under the most favorable conditions takes many times as long to warp a vessel to.

During the year 1914, active work was done on the new thousand-foot docks in New York harbor, although they were not completed. These structures are 150 feet wide and were designed to accommodate the largest of the recently constructed "ocean greyhounds" of the White Star Line, such as the *Olympic,* and the ill-fated *Titanic.* This work requires one of the largest cofferdams ever built, as it is necessary in the construction to expose the entire area where the structures are to be situated. The work of building these docks is one of the most notable tasks of its kind in the world.

The most important wet docks are located at London, Liverpool, Havre, Marseilles, Hamburg, Kiel, Calcutta, and Hongkong.

Points on Location and Design. Wet docks are best located and more necessary where: (1) rise of tide is considerable; (2) nature of trade requires ships of great length, as they are very liable to injury from taking the ground, and becoming strained; (3) bottom is hard and uneven; (4) harbor is open to entrance of surface waves of considerable height, or of a ground swell, because of the lack of protection; (5) there is ample supply of fresh water to fill the basin or dock.

In the design of a wet dock, particular attention should be given to the exposure to the sea or river, so that the gates which control the entrance may be sufficiently strong to resist the action.

Capacity. Capacity is an important factor, and may be defined as the number of vessels that can be contained within each acre. This varies with the size of craft and with the exposure, depth, etc. The form or plan of a wet dock will depend upon the nature of the ground and other local conditions; but in general it should be largest at the entrance, so as to permit of easy ingress and egress. The proportion of water area to length of quays will depend upon the shape of the dock.

DREDGING

Methods of Operation. The dredging of a channel leading to a harbor is frequently a necessary undertaking, and is performed for

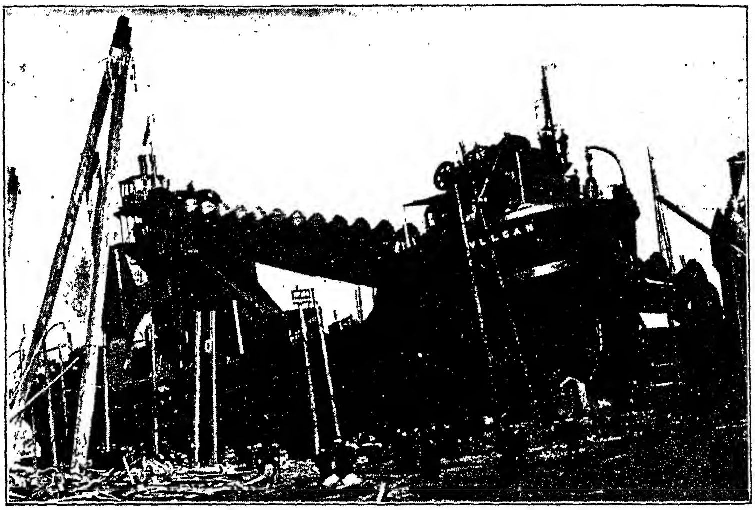


Fig. 56. Dredge of "Bucket-Ladder" Type
Built for dredging very hard material in harbor of Liverpool, England. Capacity, 2,000,000 pounds of material per hour. Works at maximum depth of 60 feet.

the purpose of increasing the depth by removing that material which has been deposited and which the ebb tide or current is unable to carry away.

In its simplest form, the operation consists merely of loosening or breaking up any such material and allowing the current to carry it away. In this connection, jets of water have been applied in various ways to remove soft material; but, as these are not particularly satisfactory, other means are usually employed.

The more general methods are: (1) to pump and discharge the soft material in an excess of water, through long pipes, to the mainland, where the silt is deposited to form "fill", and the water drains away; (2) to excavate the material by means of dredges, and load it into scows which are emptied at sea; or to load the material in the dredge itself, which, when full, is taken out to sea and unloaded.

Types of Dredges. Dredges are of various types, and will remove material ranging from the liquid state to a consistency approaching that of hard rock. The types may be divided into

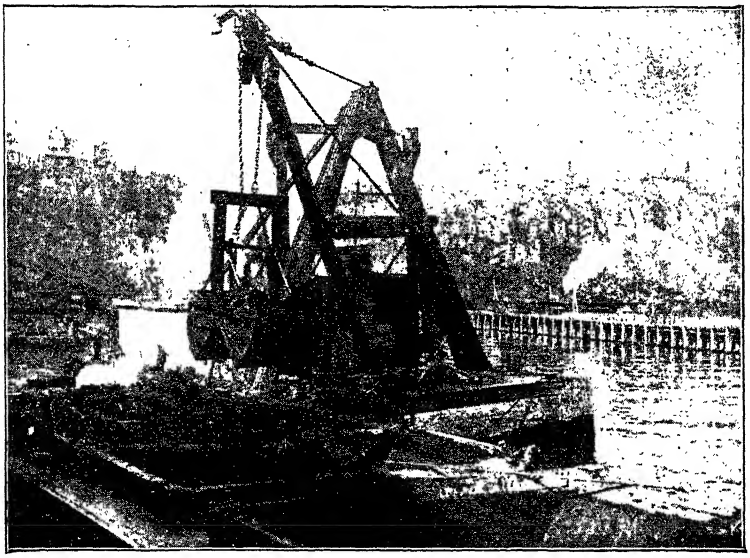


Fig. 57. Clamshell Dredge Discharging into Scow

bucket-ladder dredges, *clamshell* dredges, *orange-peel* dredges, *dipper* dredges, *pump* or *suction* dredges, and *rock-breaking* dredges.

Bucket-Ladder Dredge. The bucket-ladder or elevator dredge, Fig. 56, may be divided into two classes: those equipped with *single* ladders and those with *double*. The former can discharge on either side of the vessel; while in the case of those with the double ladders, each set of buckets can discharge only on its respective side, and consequently this type requires more room. It is maintained that the latter type works to better advantage where there is a large amount of material heavy in character to handle.

In these dredges a series of buckets attached to an endless chain

passes over revolving trunnions. The lower end of the arm to which the trunnions are fastened is lowered into the water; and the buckets, being made to revolve, excavate the material. The excavated material is carried in the buckets by the chain over the upper trunnion, where, as the buckets are inverted, it is discharged into a hopper or other receptacle, from which it is conveyed to a scow. The buckets have perforations in the bottom to allow the water to flow out. Usually the bucket ladder is situated on the center line of the dredge.

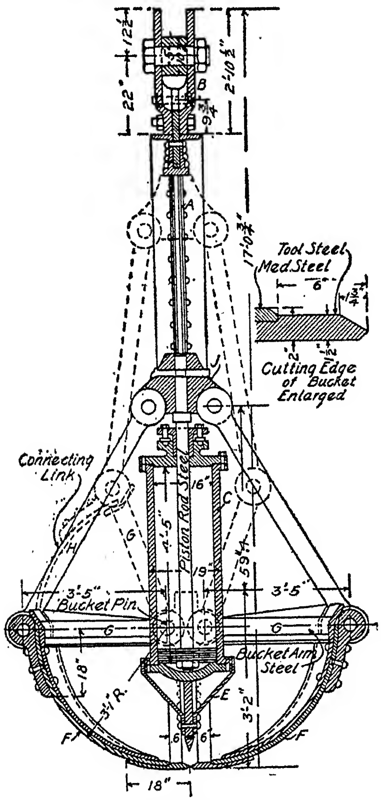


Fig. 58. Vertical Section of Pneumatic Clamshell Bucket

Capacities. These dredges will handle the hardest clay, hardpan, or soft rock; and the buckets will lift stones weighing from 500 pounds to 2 tons. A small dredge of this character, 75 feet long, 17-foot beam, and 8 feet deep, with 45-horsepower engines, can raise about 50 tons of clay per hour; while one of 250 horsepower can handle 800 tons per hour.

The largest dredge of this type is the Panama Canal dredge, *Corozal,* measuring 268 feet in length, of 45-foot beam, and drawing 17 feet. It has, attached to the ladder, 40 to 54 cubic foot buckets, and for alternate use, 34 to 40 cubic foot buckets, and will raise 1200 cubic yards in sand and mud per hour from a depth of 50 feet.

Clamshell Dredge. The clamshell dredge, Figs. 57 and 58, has a sort of hemicylindrical bucket suspended by a chain from a jib, the two halves of the bucket being kept open while it is descending, so as to dig into the material to be excavated. By tightening one of the chains, the bucket is closed, and then hoisted by the other to the surface. The load is dumped by slacking one chain and holding the

other, whereupon the bucket opens again as in the first position. Frequently a hemispherical bucket is used, divided into four parts; in this case the dredge is called an *orange-peel* or *grapple* dredge. Clamshell and orange-peel dredges are more serviceable than other forms for general use.

These dredges will operate very satisfactorily in soft or loose material; and when hard material is encountered, teeth may be used the more readily to penetrate it. In hard clay, the ordinary clamshell possesses little value.

Capacities. Clamshell and orange-peel buckets vary in capacity from 500 pounds up to 15 tons, which is the size of one used in the

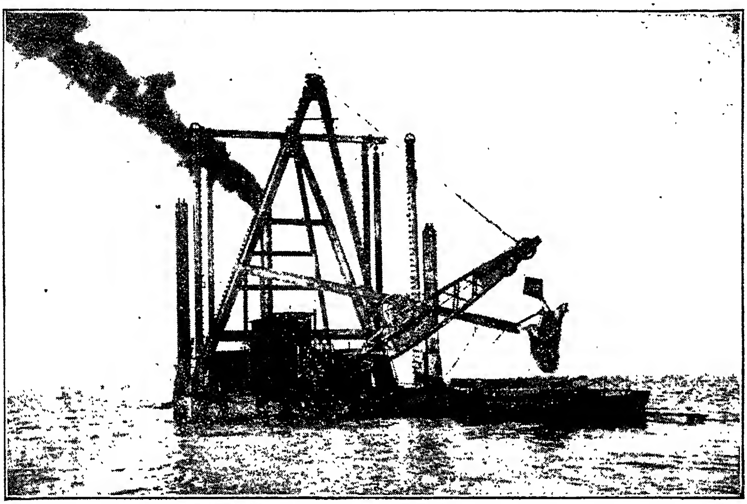


Fig. 59. Dipper Dredge Discharging into Scow

harbor at Buffalo, New York. The one mentioned operated in 65 feet of water, and has a jib 65 feet long. This type is particularly advantageous in confined places and in almost any depth of water. They may be worked by either a single or a double chain, the action being as follows: On the bucket being released, the chain runs over the pulley at the jib head, and the bucket descends through the water into the material to be lifted, with its cutting edges open, and the faces or teeth falling perpendicularly. The weight of the bucket forces them into the material. The chain is then drawn in by the drum of the winch, closing the grab and raising it.

Dipper Dredge. The dipper dredge, Fig. 59, consists of a massive steel dipper or bucket, which is moved by means of an arm, in a vertical plane, and which may be revolved on a table through a considerable arc of a horizontal circle. It is similar to the steam shovel in every respect. The process is to scrape the dipper, at the end of its arm, over the bottom of the channel, until the dipper is full, then to raise it above the water by the arm, move it horizontally through an arc of a circle by the turning of the table, open the bottom, and discharge the contents into a scow. The scow is held in position by means of timbers driven into the bed of the harbor.

Capacities. The dippers are made of sizes from $\frac{1}{2}$ cubic yard to 15 cubic yards in capacity—which latter was the size of the one used on the Panama Canal in 1914—and will operate in water up to 50 feet in depth. They have the disadvantage of requiring considerable space in which to operate, and in this respect are inferior to the clamshell type in confined situations. They, however, will excavate almost any material except solid rock.

Suction Dredge. *Pump* or suction dredges are principally used for raising sand or mud, the material being pumped up through suction tubes, and discharged either into a hopper or through pipes or troughs onto the shore. They can also be made to remove clay by having a cutter placed at the end of a shaft working in the suction tube; the cutter breaks up the material sufficiently small for detached pieces to be drawn up the suction tube. When working in heavy sand, the material left in the hopper represents about 50 per cent of the water and sand raised; in clayey sand, about 40 per cent remains; and in sticky clay, about 10 per cent.

The advantage of these dredges is that they can work in any ordinary weather, and at times when it would be quite impossible for those of other types to do so; and the material can be delivered directly for distances of from 200 to 300 yards by the force of the pump, so that silt raised from the bed of a river or a harbor may be delivered directly at the back of training walls, or used for raising land for harbor purposes. The pumps used will raise and pass through them stones or other hard substances of any size that will pass through the suction pipes.

Notable Operation at New York Harbor. Of the more recently constructed dredges of this class, the two built in 1900 for removing

40,000,000 cubic yards of material from the East channel in New York harbor, are perhaps the most notable. Their hulls were 300 feet long, beam $52\frac{1}{2}$ feet, depth 25 feet, and hopper capacity 2800 cubic yards.

When dredging in clay, the suction pipe was placed in a vertical position, and was made to travel around the end of the vessel. Attached to the pipe was a vertical shaft with horizontal cutters at the bottom, which were hooded over. As the material was broken up, it was drawn into the suction pipe by the action of the pumps. The suction pipe, made in three telescopic lengths, 21 inches in diameter, was capable of being used in 26 feet of water. The centrifugal pump could raise 16,000 cubic feet of water and sludge a minute, sometimes even bricks, stones, iron, etc., weighing from 6 pounds to 8 pounds being drawn up through the suction pipes without injury to the pumps. The material thus raised was discharged through iron pipes varying in length from 1000 feet to 4000 feet, onto low swampy ground which was thus reclaimed.

To moor these dredges, piles fitted with heavy iron shoes were suspended vertically from frames on the deck, the lower end being kept in place by guide rings. When the vessel was to be moored, the piles were set free, dropped onto the ground and penetrated it at the bottom of the channel. When the vessel required warping forward, the piles were lifted to their former position by chains passing over winches worked by the engines.

In the improvements carried out for deepening the channel leading to the harbor of New York, nearly 5,000,000 cubic yards of material were removed, which had to be conveyed $10\frac{1}{2}$ miles to sea. The quantity on which the contractor was paid was the quantity in the barges; but this was about 27 per cent less than the actual quantity removed, the remainder being carried away in suspension by the current. The material had to be raised from a depth varying from 24 feet to 35 feet under water, the total lift being between 36 and 46 feet. The excavated material consisted of mud, clay, and sand; and part of this being nearly of the same specific gravity as the water, only a certain quantity settled in the barges, the remainder going overboard with the water and being carried away by the current.

The plant provided for doing the principal part of the work consisted of 3 seagoing dredging steamers, 4 large barges, and 4 steam

tugs. These dredges varied in length from 132 feet to 157 feet, 31- to 37-foot beam, and 8- to 16-foot depth, their carrying capacity varying between 275 and 650 cubic yards. Each dredge was provided with two pumping outfits independently arranged, and each pump was capable of lifting 4200 gallons per minute. The suction pipes, 15 inches to 18 inches in diameter and 60 feet long, were located one on each side of the steamer amidships. To render them flexible so as to accommodate them to the rolling and pitching motion, about 12 feet consisted of rubber supported by chains against the vertical strain caused by the weight of the remainder of the pipes. The pipes were provided with scrapers at the bottom for loosening the material. The hoppers were divided into compartments surrounded by longitudinal sluiceways extending either way from a central receiving hopper. These sluices were provided along their source with a series of adjustable bottom and side gates, to regulate the discharge of the material into the different compartments, and prevent the listing of the vessels. The steamers worked in all but the roughest weather, and were kept under headway from the time they left their berths in the morning until they returned to them at night. On arriving at the station, the pipes were lowered, and the vessel kept constantly under steering headway until the hoppers were full, when the pipes were hoisted, and the vessel under full steam proceeded to the discharging grounds.

Rock=Breaking Dredge. Heretofore, where it was desired to remove rock from beneath the surface of the water, various means of applying powder or dynamite to break up the material first were employed, after which some sort of dredge would remove the broken pieces. This has been replaced to some extent—at least where the rock is soft—by using strong steel claws fastened to the buckets; or every other bucket has been replaced by them.

In the case of very hard rock, heavy chisels have been employed, which, falling upon the material, crush it by their weight, after which it may be removed. The height of the fall varies from 10 feet to 20 feet; and the weight of the chisel or ram, from 4 tons to 10 tons.

Several machines like this were used for removing a shoal containing about 3,000,000 tons of hard limestone rock in the Suez Canal. The first dredge was 180 feet long, 40 feet beam, and 12 feet draft. It was fitted with 10 heavy chisel-pointed rams or rock

cutters, each 42 feet long and weighing 4 tons. Five of these were fitted on each side of the well through which the broken rock was lifted by a ladder and buckets, the bag of the bucket chain being supported by a guide wheel specially designed to relieve the strain on the bearings, bucket links, and pins. These rams were raised by hydraulic power, and allowed to fall from 10 feet to 20 feet on the rock, delivering between 200 and 300 blows an hour. The bucket chains were driven by a 4-cylinder compound engine of 200 horsepower. When at work, the machine was moved over the surface in a series of arcs by winch motion arranged by swinging the vessel from side to side, pivoting on a steel mooring pile 3 feet in diameter, which passed down through the hull in the after part of the vessel and rested on the rock. There were two of these piles maneuvered by hydraulic power, enabling the vessel to advance a given distance after each swing, and preventing her losing her position. Dredges somewhat similar to the above are being used at present by the United States Government on the Panama Canal construction.

ATLANTIC COAST HARBOR IMPROVEMENTS

Study of Sand-Movement Forces. The improvement of harbors on that part of the Atlantic coast south of New England involves almost entirely such protective or corrective works as are necessary for a sandy coast line. In making a study of these works, it is necessary to consider forces which cause the movement of sand; for almost the whole object of river and harbor improvements on a coast line of this character is the control of the movements, or the retention in place, of large masses of sand that would otherwise be prejudicially shifted by winds and waves or tidal currents.

Wherever dry sands are found, they are always shifted by prevailing winds; indeed sand is drifted about under the action of winds precisely like snow. Captain Black, in his paper on the improvement of harbors on the south Atlantic coast of the United States, asserts that on the Florida coast sand moves like drifting snow with a depth of from 4 inches to 8 inches, while a properly constructed sand catch will form a dune 10 feet high in a single year. It is thus evident that, inasmuch as the sand will be moved in the direction of the wind, if that direction is across a channel the result will be

deposition of sand in that channel, with a possible obstruction of the latter so far as the passage of vessels of considerable size is concerned. Similar results from the same wind forces are constantly produced on those portions of the Pacific Coast which are sandy.

Waves and Currents. The most troublesome forces to control in connection with the littoral movements of sand, are those of waves and currents—particularly tidal currents, which are largely shifted in direction by bars and shoals of their own creation, though the winds, also, frequently create difficulty. The study of wave motion has been largely developed by Scott-Russell, Hagen, the Weber Brothers, Rankin, Froude, and others, but it is not necessary to consider in this connection the analytic basis of such work. It is only requisite to observe that wave motion is a combination of rotation and translation. Stevenson, in his work on rivers and harbors, states that the height of the highest point of a wave crest above the normal level of the water in which it is produced is two-thirds of the wave height from hollow to crest, and that the velocity of wave translation in the Atlantic Ocean may reach as high as 115 feet per second, while ordinary storm waves travel at a rate of 50 feet or 60 feet per second.

It is a matter of common observation that waves break as they roll in toward the shore in shallow water; and Stevenson states that the mean depth at which they will commence to break is from once to twice a wave height, but that they break earlier when the water shoals rapidly than when the slope of the bottom is more gradual. It is, of course, perfectly evident that the wind will have great influence upon the velocity of wave translation, as well as upon the shape of the wave and the circumstances under which its crest will break. A sufficiently high wind will cause almost any wave to break at its crest, and will cause each molecule to describe a very different orbit from that assumed in mathematical analysis.

It is evident that the crest of a wave will be at right angles to the direction of the wind, and that its motion will be along that direction; but when adjacent to the shore line, if the wind is blowing obliquely to it, one end of the crest will be retarded and deflected by the shore, while the other will be in deep water. Consequently the crest will swing around more or less toward the shore line, and produce complicated motions in the water.

Back Flow. As a wave rolls inshore, the water is piled up on the beach for a moment and then flows back; and inasmuch as successive waves are rolling in along the water surface, its back flow will result in an undercurrent along the bottom at a constantly decreasing rate in receding from that part of the beach at the low-water line. The motion of the waves will produce great disturbances of the bottom in shallows; in fact it will cause the constant picking up of sand or sediment which will be carried inshore by inshore winds and by the motion of translation of the waves, until eventually the material is piled upon the beach. The back flow or undercurrent on the bottom will, on the other hand, tend to carry material deposited on the beach out to deep water, but with very much less activity than the inshore forces just mentioned. As a consequence, the beach constantly gains material; and inasmuch as the back flow or undercurrent is relatively weak, the finer portions only will be transported back toward deeper water. This accounts for the clean and relatively granular character of most sand beaches.

A current of water which is too weak to erode a bottom will transport very considerable material in suspension or in semi-suspension if it can be once placed in that current. Various authorities name currents from 6 inches to 15 inches per second as sufficient to begin a scour upon a sandy bottom of various degrees of coarseness. The liability to scour, however, of any sandy bottom, depends to a considerable extent upon the degree of solidity with which the sand is packed. Captain W. M. Black, in the paper already cited, states that he has seen a bank of packed sand remain without erosion under a current of 3 feet per second. With a relatively high current, considerable material will, of course, be transported, while in the case of a slower current the amount will be considerably less. Hence, if the back flow or undercurrent on a beach is relatively strong, considerable material will be carried into the water from the beach, and the resulting bottom slope will be much more gradual than if the back flow is weak and able to transport but little material, and that of the finest.

If the shore is eroded by a littoral current, the sand which is transported by the latter will be deposited with the decrease of velocity which takes place as the current flows into deeper water, and a spit or shoal point will thus be formed. Again, if these currents

are formed and directed by the winds, this deposition of sand may become extremely variable and form hooks or loops, thus producing a shoal bottom of eccentric character; at the same time these sand formations in the shape of bars or shoals may be spread across a river entrance or harbor.

It is believed from many observations, that sand is much more in motion or "alive" during flood tides than during the ebb, although some perfectly reliable observations to the contrary have occasionally been made. It is known that a new beach formed during a gale will at first be comparatively soft, and eventually become very hard, under the pounding action of breakers; in fact, some beaches become almost as hard as asphalt pavements, so that they will ring under a blow from the foot of a horse.

Wave Action on Jetties. The force of waves on the Atlantic Coast of the United States is somewhat less than that of similar action upon some foreign shores, although they are seen frequently to exert destructive action upon protective works. The following accounts are taken from the Report of the Chief of Engineers, U. S. A. of 1890, on the Improvement of St. Augustine Harbor, as well as from the paper of Captain Black already cited:

A wave may act on the jetty directly, either by a blow or a push, or a blow and a push combined; and indirectly, by a pull, by compressing the air in the voids of the masonry, by upward pressure due to the difference of head produced on the two sides of the jetty, or by a combination of these actions. The direct action measured on the dynamometers had effects equal to pressures varying between 190 and 753 pounds per square foot. This action took place when a wave broke directly on or in advance of the jetty; this also compressed the air in the voids of the jetty. . . . . . Jets of water and sand were sometimes projected up from the cracks in the jetty to some height. The maximum height of any wave observed striking the work was 6 feet. . . . . Up to a height of about 2 feet above mean low water, riprap weighing 40 to 50 pounds was but little disturbed. Above this limit, to the height of 10 feet, the highest point observed, riprap varying in weight from 40 to 200 pounds, could not be held at any slope. An isolated piece of concrete weighing 350 pounds, and resting on its flat base 1.7 square feet in area, with its center of gravity 7 feet above mean low water, was moved several feet by breakers whose crests were about $7\frac{1}{2}$ feet above mean low water. These breakers measured $3\frac{1}{2}$ feet from hollow to crest. . . . . . All that portion of a mound or wing composed of riprap (varying in weight from 40 to 220 pounds) tightly chinked with oyster shells, lying between 4 and 6 feet above mean low water, no matter what side slopes the riprap was given, would be carried away in a single tide whenever breakers greater than 4 feet in height struck it fairly. . . . . . A block of concrete weighing 527 pounds was elevated 1.3 feet by the action of

a single breaker. During the same tide, it was moved 23 feet inshore. A dynamometer within 8 feet of its original position recorded a maximum pressure of 575 pounds per square foot during this tide. A piece of concrete weighing 200 pounds was lifted vertically to a higher level than that of the water surface, by a wave which broke just in front of it. Another block of concrete weighing 1600 pounds was lifted from its bed vertically at least 14 inches, and then moved several yards. Later, a concrete block 10 by 6 by $2\frac{1}{2}$ feet and weighing dry 21,000 pounds, lying about at the mean low-water line of the beach, was lifted vertically 3 inches, and there caught and held fast. The maximum wave height and dynamometer readings during that gale were 5.5 feet and 633 pounds, respectively.

Coast from North Carolina to Florida. The coast from Cape Hatteras to the southern extremity of Florida is low and sandy, broken by many openings into rivers, bays, and inlets. The highest point in the entire line is Mount Cornelia, at the mouth of St. John's River, and it is only 60 feet high. There is a general motion toward the south, of the sand on this entire stretch of coast, in consequence of the direction of the prevailing winds, which is in the main southerly.

The slope of the country bordering the coast is invariably very slight; consequently all the rivers emptying into the ocean through it are tidal streams, the effect of the tides being felt for a considerable number of miles back from their mouths. As a result, the material brought down by the different streams does not reach the ocean, but is deposited in the basins immediately back of their mouths, except in a very few cases where small amounts may reach the ocean. The slope of the beach along low-water line is also very slight, being but 1:45 off Cape Hatteras, and as low as 1:1000 at Sapelo Inlet just south of Savannah, from which point it increases southward until it reaches 1:55 at Cape Canaveral, about midway down the coast of Florida. The decrease in bottom slope in the vicinity of Florida is largely due to the bay-like shape of the shore between Cape Hatteras and Cape Florida. As the tides move into this "Southern Bay", as it is called, they become concentrated in their effects, and rise in height as the point of the bay in the vicinity of Savannah is reached. The material moved by the tides is therefore concentrated in the same vicinity, and a steeper slope of bottom results.

In the vicinity of Cape Hatteras, the prevailing winds are from the north and northeast, and the same is true for the vicinity of Charleston, while on the coast of Florida the reverse condition takes place. Nevertheless, in all cases, the heaviest storms and highest

winds are from a northerly direction, which produce at all points along the coast the southerly movement of the coast sand, which has already been mentioned. It is important to regard this fact in the designing and construction of correction works at the mouths of the more important rivers; and it will be noticed in many cases that this tendency has deflected the mouths of many of the rivers in a southerly direction.

Summary of Conditions. The general conditions prevailing on the portion of the coast under discussion may be summarized thus: (1) shores are low, fringed by a beach of light, shifting sand, the foreshore having a very gentle slope, giving comparative protection from wave action; (2) heaviest and most prolonged gales are from the northeast, with occasional severe gales from the southeast; (3) mean tidal range is from 1.5 feet to 7 feet, with two tides daily; (4) there are no continuous, strong littoral currents, and no strong tidal currents excepting in the immediate vicinity of the openings on the coastline; (5) coast storm drift of sand is large in volume, with a resultant movement to the south; (6) this drift forms fan-shaped shoals across all openings on the coastline; (7) there is little or no fresh-water sediment brought to the coastline; and (8) channels across the shoals are generally shifting in character; while the navigable channels are maintained by the ebb outflow from the interior tidal basins, reinforced by the fresh-water discharge from the interior.

Improvement at Cape Fear River Mouth. The city of Wilmington, North Carolina, is situated nearly 28 miles from the mouth of Cape Fear River; and for about one-half the distance, the latter is roughly parallel to the sea beach, from which it is separated in the main by a narrow stretch or tongue of sand, in some places but a few hundred feet wide, but, near its mouth, by the accretion of sediment called "Smith's Island", which is 2 miles wide and intersected by numerous threads of water. It is probable that in the early age of the coast, Cape Fear River emptied into the ocean at a point nearly opposite Wilmington, and that the southern motion of the coast sand gradually deflected its mouth by slow stages to its present position.

There are historical reasons for believing that Cape Fear River had a channel capable of passing vessels of 14-foot draft through Bald Head channel to the city of Wilmington, up to 1761; but in that

year a violent equinoctial storm of 4 days' duration caused the sea to break through the narrow sand beach at what is now known as New Inlet, some 8 miles up the river from the former entrance. This new channel was reported at different times to have a depth of from $2\frac{1}{2}$ feet to 18 feet; but the effect of this shortened route to the sea for the river water was a scouring from a certain depth of 6 feet at low water in 1797 to 10 feet at low water in 1839, with a corresponding shallowing by bar formation in both the Western and Bald Head channels, although up to 1839 the latter was the main entrance to the river.

From 1839 to 1872, that channel was discontinued, while New Inlet and the Western channel were used as the main entrances to the river. Inasmuch as New Inlet was vastly more exposed to the prevailing northeast winds than the natural or Bald Head channel, making both the entrance for vessels and the preservation of correction works much more expensive at the former point, it was determined in 1872 to close New Inlet and reopen and maintain Bald Head channel. In consequence of the decreased velocity in the old channel, due to the efflux of water through New Inlet, there were produced horseshoe shoals and a fan-like bar formation at the outer extremity of Bald Head channel. The shoaling influence at different points was also considerably complicated by the tidal impulse up the river through Bald Head channel, not being felt at New Inlet until considerably later than on the sea front.

New Inlet Dike. The New Inlet dike was constructed between 1874 and 1881. It was begun by constructing a cribwork pier 500 feet long out from Federal Point for the usual pier purposes for such work. The same scouring effect around the advancing end of the cribwork was experienced which gave the same kind of trouble in the construction of the Zekes Island dike, and resulted in deepening the water from 6 feet to 12 feet at a point 200 feet from the shore end of the pier. The pier was completed in the month of November, and the end crib was sunk in 19 feet of water, where the original depth was but 6 feet. The settlement of these cribs and the extreme difficulty in building upon them fast enough to keep them above low water, caused their abandonment for all subsequent work. In 1875 it was determined therefore to close the remainder of the New Inlet opening by first sinking an apron of log mattresses cov-

ered with a layer of brush and loaded with stone, in which the logs were laid normal to the axis of the jetty. The scour in front of the advancing end of the apron did not exceed 3 feet, and there was no settlement after the mattresses were placed in position.

The jetty was completed by dumping one-man riprap stone upon the apron already in position. The slope of the finished work on the sea side was about 1:2, and on the land side 1:1½. On the top of this mass of riprap and above low water, there was carefully laid a capping of undressed granite blocks weighing from ½ ton to 4 tons each, with a slope of 1:2 on the sea face, and a slope somewhat less on the land slide. While this system of construction was an improvement upon the crib plan, it was found that the logs did not close the bottom of the jetty in a perfectly water-tight manner; hence material was gradually scoured out between the logs, causing them to settle. No settlement, however, was found where the original depth was greater than 13 feet below low water; but in some places there was a scour of 8 feet to 12 feet along the edges of the foundation, due to the flow of water over the top of the dike. In consequence of the irregularity of the tide effects at this point, the difference in head of water on the two sides of the dike was as much as 2 feet at some stages of the tide. This difference in head, combined with the flow over the top of the dike, caused the scour between the logs.

The crest of the completed dike is 1 foot above high water of ordinary spring tides, and the total length of the entire work is 4752 feet; it contains nearly 183,000 cubic yards of material; and it cost complete about $2.75 per cubic yard, or $105 per linear foot. This completed work joined the Zekes Island dam at its northernmost point, and completely closed the opening between Federal Point and the northern extremity of Smith's Island.

Dike at Bald Head Channel Mouth. The only remaining jetty work for the improvement of this river mouth, is the dike across the shoals at the mouth of Bald Head channel, which has a length of 5200 feet, and was built between 1883 and 1888.

In consequence of the experience with the log foundation of the mattresses already described, it was determined to employ a different plan for this dike of mattresses. The foundation consisted of a brush mattress apron, part in sections and part continuous, in

5 feet to 12 feet of water. This mattress was broad enough to extend beyond the side slopes of the dike, and thus to act as an apron for overflow. It consisted of three layers of brush, of which the lowest was transverse to the axis of the jetty, the next longitudinal, and the third transverse. These were compressed and held by fascine copes, or binders, 5 inches to 8 inches in diameter, strongly bound with spun yarn. The binders were parallel to the axes of the jetty, and were spaced $3\frac{1}{2}$ feet or 4 feet apart in sets of two, one above and one beneath the mattress. The brush was compressed between the binders by a lever to the full strength of the 9- to 18-thread ratlin which tied the upper and lower binders together. The average thickness of the mattresses after compression was 9 inches. They are described as strong and pliable.

A portion of the foundation was made of separate mattresses, constructed on a tilting table, 90 feet by 30 feet, and towed to the site of the work and sunk. A length of 5200 feet of the foundation was made of continuous mattress work in two parts of 3100 feet and 2100 feet, on a scow 80 feet by 28 feet, provided with an inclined table 60 feet by 36 feet. A section having been launched, its binders were spliced to binders on the scow; another section was made, and the scow run from beneath on the line of the axis as before. The sinking of the mattress by stone followed a short distance behind the scow. In the small mattresses made on the tilting table, it was found that by practice an average of 25 feet in length of fascine binders was made each hour by one man. After the fascines were ready and the binders spliced, the mattresses were made and fully bound at the rate of $7\frac{1}{2}$ square yards for each hour's labor.

The dike was finished in 1888 with a total amount of 76,000 cubic yards of stone used in the construction, and a crest 6 feet wide at the level of high water of ordinary spring tides. The side slopes are $1:1\frac{1}{2}$. The crests and slopes to low water are capped with heavy stone, hand-laid to a smooth face, with an average thickness of from 9 inches to 12 inches. The average subsidence of the dam due to settling in the mud, compression of mattresses, and consolidation of riprap, was about 2 feet.

It was necessary to dredge channels both through Horseshoe Shoal and the short shoal about $1\frac{1}{2}$ miles to the north of it, as well as through the outer bar along Bald Head channel. The dredge

used for this work was of the hydraulic type. She was a small propeller of 145 tons' burden, and was able to work very satisfactorily on the outer bar, except in stormy weather, when she was usually operated on the shoals inside the river mouth. The excavation was made with a 9-inch centrifugal pump driven by a double oscillating engine with 10-inch cylinders and with two suction-pipes of wrought iron, of $6\frac{3}{4}$ inches internal diameter. She was able to dredge in a depth of water from 6 feet to 14 feet, and frequently raised her bin capacity of 45 cubic yards in 25 minutes; and the average day's work was 515 cubic yards.

Lesson of Experience. The experience gained in this work all shows in a marked manner the inadvisability of attempting to force a dike or a jetty of full depth across a channel with a sandy and shifting bottom, in water in which there is an appreciable current. An increasing depth of scour will invariably take place in front of the advancing point of the work, and the difficulties caused both by construction and maintenance will be very largely increased.

The proper plan to follow, then, is the one which has been almost universally adopted by the later and similar works, that of first sinking a pliable foundation which can be quickly put in place without greatly disturbing the flow. In this manner a bottom or foundation is secured for the superstructure which will not yield to destructive forces. It was found, however, that water would even find its way under the Bald Head channel dike to some extent, and produce a slight settlement. Some engineers have recommended that such foundations be formed of riprap which settles into the soft or yielding bottom and forms a part of it; but, on the whole, brush fascines have been found to be quite satisfactory.

The result of all this Cape Fear River improvement work has been a channel with a low-water depth of 17 feet through the outer bar in Bald Head channel, and 16 feet through Horseshoe Shoal, the dredged channels having a width of 200 feet, with the depths named.

REVIEW QUESTIONS

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART I

---

1. Define the terms *work*, *power*, and *energy*. What are their units of measurement?

2. Illustrate the relation between *pressure-head*, *velocity-head*, and *gravity-head*.

3. Define the terms *absolute velocity* and *relative velocity*.

4. Distinguish *impulse* and *reaction; static pressure* and *dynamic pressure*. To give an impulse of 120 lb., what must be the velocity of a jet of water issuing from a nozzle 1.25 inches in diameter?

5. Explain how the dynamic pressure of a jet of water can be measured.

6. What, in general, is the difference between a water wheel and a turbine?

7. Discuss the requirements for high efficiency in hydraulic motors. To what causes are losses of energy due?

8. Compare the respective advantages or disadvantages of, overshot, breast, and undershot wheels.

9. Illustrate by diagrams, and discuss the significance of, different shapes of buckets on overshot wheels.

10. What conditions should be observed in installing overshot, breast, and undershot wheels?

11. Distinguish several different types of impulse wheels.

12. Describe the essential features of the downward-flow impulse wheel.

13. Compare the respective advantages of inward and outward flow in the case of impulse wheel.

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART II

1. What are the fundamental parts of a turbine?

2. Explain the various ways in which turbines are classified, describing briefly the main characteristics of each type.

3. What are the most common types of reaction turbines used in America?

4. What is the American vortex type of turbine?

5. What type or types of water motor should be employed for low, for medium, and for high heads of water?

6. Under what circumstances is it advisable to use turbines with horizontal shafts? With vertical shafts?

7. Describe the various methods of estimating the quantity of water delivered to a motor, stating under what conditions each should be used.

8. How is the amount of leakage determined, and what is its relation to the quantity of water delivered to the motor?

9. Describe in detail what is meant by the term *effective head* on a turbine; on an impulse wheel; on an undershot wheel.

10. What is the object of testing a hydraulic motor?

11. Describe in outline the main features comprising the complete testing of a turbine by the friction brake or power dynamometer method, using whatever diagrams may be necessary for the purpose.

12. Explain how the values were obtained in columns 2 to 7 inclusive, of the table showing results of test of Tremont turbine. What general conclusions may be reached by the examination of such a table?

13. Using the appropriate formulas, discuss the influence of the approach angle α, and the exit angle β, on the efficiency of an impulse and of a reaction turbine. What general conditions must be fulfilled to insure high efficiency in a turbine?

14. Define the terms *gate-opening; part-gate; clearance; complete* and *partial admission.*

15. Contrast the main characteristics of impulse turbines with those of reaction turbines, giving their relative advantages and disadvantages under different conditions.

16. Ordinarily a reaction turbine operates with lowered efficiency at part gate. How may this disadvantage be obviated?

17. What is a *draft tube?* Describe its action.

18. How may a draft tube be used in connection with a Pelton wheel?

19. Describe the ordinary types of gates for regulating turbine flow.

20. Describe the general construction and the function of a *thrust* or *balancing piston;* a *thrust chamber.*

21. Where should the following devices be placed, and what are their respective functions: *stop valves; air valves; blow-off valves; automatic stop valves; gages; relief valves; stand-pipes; air chambers?*

22. How is the speed regulation or governing of turbines accomplished? What is *over-governing* or *racing;* and how is it prevented?

23. Describe the general features of a *headrace* and a *tail-race.* What precautions must be taken in the case of sand-bearing waters?

24. What are *water racks,* and where should they be located?

25. What is the function of a *head gate?* Where should it be placed? What different types of head gate are in use?

26. What is a *penstock,* and of what materials may it be constructed? What precautions should be observed in regard to the velocity of the water in the penstock, and in regard to change of direction?

27. How may a long penstock be protected against injury from water hammer?

28. What special precautions are necessary in the case of a penstock one section of which is nearly horizontal, with an adjoining section located on a steep grade?

29. What are *expansion joints?* When and where should they be employed?

30. What are the advantages of a wood-stave penstock?

REVIEW QUESTIONS

ON THE SUBJECT OF

WATER-POWER DEVELOPMENT

PART III

1. Name the principal parts of a typical hydroelectric power plant.

2. What devices are used for preventing floating ice and débris from entering turbine chambers?

3. What precautions are to be observed in the construction of headraces and tailraces?

4. How would you attack the problem raised by the headrace water carrying considerable quantities of sand in suspension? Is the removal of the sand in all cases advisable?

5. What rules should be observed in determining the size of conductors for conveying water to and from turbines?

6. What mechanical devices are employed for regulating the speed of water in penstocks?

7. Is the length of the penstock ever an unimportant factor? Explain.

8. Name, and describe briefly, the different types of head gates for turbine plants, and the mechanism for operating them.

9. Enumerate the principal losses of head of water which occur between the headrace and the tailrace. Indicate briefly how these losses may be minimized.

10. Under what conditions is high speed of water in penstocks permissible?

11. Under what conditions is low speed of water in penstocks advisable?

12. Of what materials are penstocks composed? Compare their relative advantages.

13. Describe briefly the important difference in the general features of design between the installation of the Hydraulic Power Company, and those of the Niagara Falls Power Company.

14. Describe the main differences between the two installations of the Niagara Falls Power Company.

15. In what respect does the Hydraulic Power Company's power development resemble that of the Ontario Power Company?

16. What are the most noticeable features in the design of the Snoqualmie Falls Power Company's development?

17. What is the greatest head of water under which any power plant in the world is operated, and what kind of motors are used?

18. What power plant is operated under the greatest head of water in the United States, and what type of motor is used?

19. Define the terms *peak load; load factor.*

20. In what different ways may peak load be provided for?

21. How is the load factor determined in any given case?

22. Describe in outline an equitable method for determining the rate of charging, and also the total monthly charge, for power, when the minimum and maximum rates per horsepower-year are fixed.

23. Explain how to change cost per horsepower-year into cost per kilowatt-hour.

24. What may be considered reasonable figures for the cost of water-power production under different conditions? What may be considered reasonable charges to the consumer for water power under varying conditions?

REVIEW QUESTIONS

ON THE SUBJECT OF

RIVER AND HARBOR IMPROVEMENT

---

1. What is meant by the *regimen* of a river?

2. What is the object sought in the improvement of a river?

3. State what you understand to be the purpose for which a river survey is undertaken, and of what it consists.

4. What is the *watershed, topography, hydrography,* of a river basin?

5. How does the vegetation on a watershed affect the *run-off,* and what is meant by the latter term?

6. What is the purpose of determining the flow of a river?

7. What methods may be employed in gaging a stream or the flow of a river? Discuss fully.

8. Discuss the formula $V = c\sqrt{rs}$, stating what each factor means, and how c is obtained.

9. What is meant by *mean velocity?*

10. Discuss the various forms of floats used in measuring the flow of a river.

11. Discuss fully how the subject of floods enters into the question of the improvement of a river.

12. What methods have been suggested as a means of protecting adjoining land against floods?

13. Discuss *cut-off.*

14. What is meant by *diversion of tributaries?*

15. Discuss fully the effect of storage reservoirs upon stream flow, taking up the subject of natural and artificial reservoirs.

16. Show the effect that the Great Lakes have upon the flow of the St. Lawrence River.

17. What are *artificial outlets,* and how may they be employed in river improvement?

18. What are *levees?* For what purpose, and how are they constructed?

19. What is the purpose of river bank protection, and how is it secured?

20. What are *revetments*, *mattresses*, *fascines*, and how constructed?

21. What are *dikes?*

22. Discuss *erosion* and *transportation*.

23. What is the purpose of a *commercial harbor?* How does a *harbor of refuge* differ from the above?

24. Discuss winds, waves, currents, and tides.

25. State the types of breakwaters.

26. Of what may breakwaters of the vertical type be constructed? Of what may those of the mound type be built?

27. Discuss each form of construction of each type, and its advantages or disadvantages.

28. What regulations are in force in the British Admiralty in regard to lighting and buoying estuaries and channels?

29. What forms of buoys are used?

30. Discuss the relative efficiency of the different illuminants used in lighting rivers and harbors.

31. Give a brief account of the Eddystone Lighthouse.

32. What general rules should be observed in the construction of lighthouses?

33. What are *dry docks?*

34. What are *wet docks?*

35. What methods are employed in dredging?

36. Compare bucket-ladder, clamshell, orange-peel, dipper, suction, and rock-breaking dredges.

37. Discuss the harbor improvements on the Atlantic Coast.

INDEX

The page numbers of this volume will be found at the bottom of the pages; the numbers at the top refer only to the section.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.

Note.—For page numbers see foot of pages.